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DESIGN REQUIREMENTS FOR PICKUP AIRCRAFT

by ARTHUR B. SCHULTZ
Chief Engineer
All American Aviation, Inc.

ONE of the most fascinating pioneering jobs in aviation has been the development of the technique of picking up loads by aircraft in flight. In no other practical way can the advantages of high-speed air transport be fully realized. Further, air pickup brings these advantages to every town, farm, jungle, and isolated spot in the world without the necessity of establishing landing and take-off facilities. Nation-wide, overnight air mail, air express, and air freight will be possible. Remotely located mines and other relatively inaccessible developments hitherto uneconomical to operate may be opened up. Whole new industries may be created because of the speed, flexibility, and economy of this new means of transportation. The military and rescue value of cargo, glider, and man pickup is enormous. In size, pickup will keep pace with the largest aircraft being built or to be built. In completing scheduled air-mail operations, it has shown a 90% average over a five-year development period flown across the worst stretches of the Alleghenies in summer and winter. Last year, the average was 95.4% and, with the advent of better radio and radar, even higher reliability is anticipated.

■ Cargo Pickup

The history of this development can be traced to patents on unworkable devices dated as far back as 1913; but it was not until 1938 when the late Richard C. duPont became interested in the problem that any appreciable progress was made. At that time the station consisted of two masts, each 50 ft high, between which suspended a rope on which a small mail bag was hung. The pickup airplane flew overhead at about 100 ft, fishing for the loop with a grappling hook on a ⅜-in. diameter cable about 100 ft long. Successful pickups were only made about one out of every two tries. An equally impractical device was used to receive the mail bag being delivered. This system in its entirety was quickly discarded, however, and within a year's time the basic principles of the present pickup system were evolved.

The heart of the modern air pickup system is the energy absorption unit. The present type was developed only

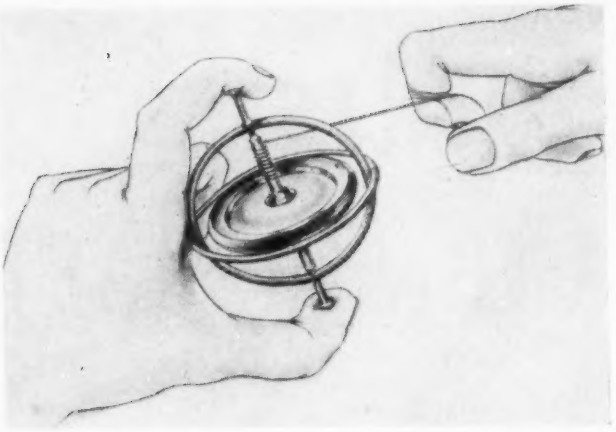
THEORETICAL principles of cargo pickup are outlined here by Mr. Schultz, who describes the design of pickup units, contact means, transfer equipment, and ground stations for both 200-lb and 60-lb units. He also outlines the basic features desirable in future pickup airplanes, as well as structural design requirements.

The author then describes glider pickup, giving consideration to the basic theory behind this type of pickup, the design of pickup units, their installation in the airplane, the cable guide system, and contact means. He outlines the design features of glider pickup airplanes and structural design requirements to be used in the installation of the equipment.

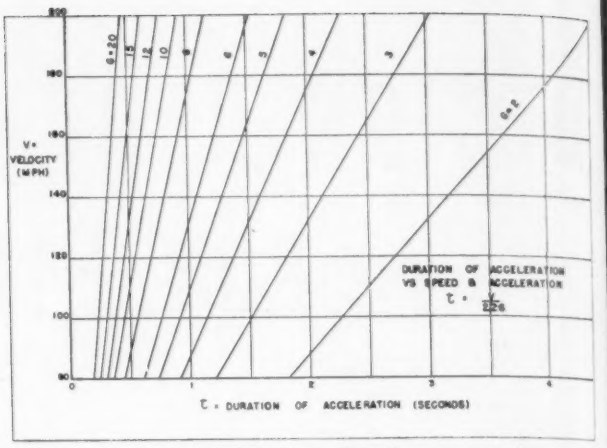
In closing the author points out that aircraft pickup has obvious military possibilities and that the peacetime potential is practically unlimited.

THE AUTHOR: ARTHUR B. SCHULTZ (M '30), designer and nationally known soaring pilot, is chief engineer of All American Aviation, Inc. The ABC intermediate sailplane, which won the Eaton Design Competition in 1937, was created by Mr. Schultz, and it earned for him the "Silver C" rating, an international award for high performance achievement. In 1941 the author became associated with the late Richard C. DuPont, and he took an active part in the design and development of glider pickup devices now used by the Army Air Forces. Mr. Schultz, who is a director of the Soaring Society of America, was formerly chief engineer of Kermath Mfg. Co.

[This paper was presented at the SAE National Air Cargo Meeting, Chicago, Dec. 6, 1944.]



■ Fig. 1 – Force and time are required to accelerate a rotating mass to high speed



■ Fig. 2 – Duration of acceleration plotted against airplane speed and rate of acceleration

after much experimentation with various devices such as hydraulic pistons and motors and several types of brakes and reels. The wrinkle that solved most of the basic troubles in pickup was the idea of using the elasticity in nylon rope to absorb the inertia of the winch drum as it is accelerated to aircraft speed.

A pickup may be described as comprising four steps:

1. Establishing a connection between the moving airplane and the stationary object.

2. Accelerating the mechanism in the airplane so that the connecting line pays out at airplane speed.

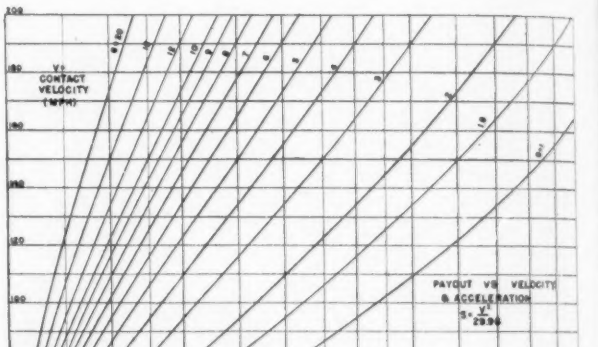
3. Braking the payout of line so as to impose an accelerating force on the object and get it up to airplane speed.

4. Reeling the burden into the airplane.

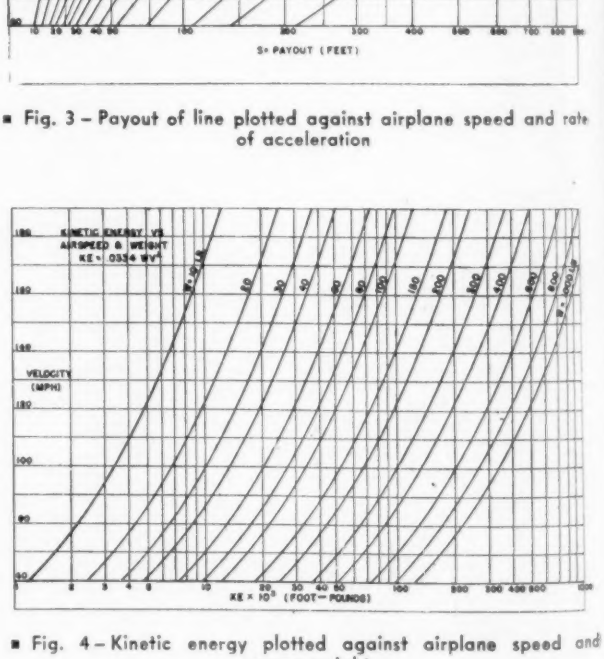
The important point lost sight of in most of the early developments was the magnitude of the forces set up in the pickup system at the instant of contact while accelerating the mechanism in the airplane. Fig. 1 is shown to bring home the fact that force and time are required to accelerate a rotating mass to high speed. In the case of air pickup this must be accomplished in less than a tenth of a second, and, since the rotating weight is several pounds, the force must be very high. In a tenth of a second an airplane moving 120 mph moves 17.6 ft and only an elastic rope like nylon has the stretch and strength necessary for the job.

After the drum has become accelerated an automatic timing device applies a brake set at a predetermined torque. As the drum comes to a stop the load is accelerated to airplane speed at any desired value from 5 to 10 or more g's depending upon weight of container, brake setting, and several other minor factors. After the pickup an electric motor winches the load into the airplane.

The rate of acceleration of the object to be picked up is controlled by the pendulum effect of the container after pickup, which will permit it to strike the ground if the climb of the pickup plane and the drag-weight ratio of the container are too low. Under conditions of human pickup, where the rate of acceleration must be carefully limited to 6g, a 3g pullout is necessary to clear the usual obstructions. In regular airline operation, where more than a 2g pullout is undesirable, the cargo should be accelerated at 10g in order to clear the ground properly.



■ Fig. 3 – Payout of line plotted against airplane speed and rate of acceleration



■ Fig. 4 – Kinetic energy plotted against airplane speed and cargo weight

The pickup is made in a surprisingly short interval of time, as may be ascertained by reference to Fig. 2. In this chart acceleration time versus air speed at various rates of acceleration in terms of g's (or times gravity) is shown. The basis of this chart is simply the formula: $v = at$. Thus, a pickup at an average of 8g at 120 mph is accomplished in only 0.7 sec.

Similarly, the approximate payout of line is determined by reference to Fig. 3, where payout versus speed at various g's is shown. The basis of this chart is simply: $s = \frac{1}{2} at^2$. In the above case the payout is approximately 65 ft.

As pickup units get larger and the thermal capacity of the braking elements becomes increasingly important, Fig. 4 will become useful. Here, kinetic energy is plotted against air speed for various weights. It can be shown that the heat absorbed in the brake of the pickup unit is approximately equal to the kinetic energy imparted to the object picked up. The formula is basically: $KE = \frac{1}{2}Mv^2$ For example, a 200-lb object at 120 mph imparts 96,000 ft-lb of kinetic energy to the brake. The cargo pickup units contemplated for the immediate future have ample reserve thermal capacity.

Delivery of cargo is accomplished either by dropping the load to the ground in a special container from a low altitude just prior to making the pickup or else at a few hundred feet altitude by the use of parachutes. The former method has been successfully used for the past five years on mail pickup operations. The latter is used only for fragile cargo as it involves a second pass over the station at low altitude to make the pickup, thus cutting down the airplane's block-to-block speed. Improvements are being made as the needs arise and considerable research in this direction is under way at the present time.

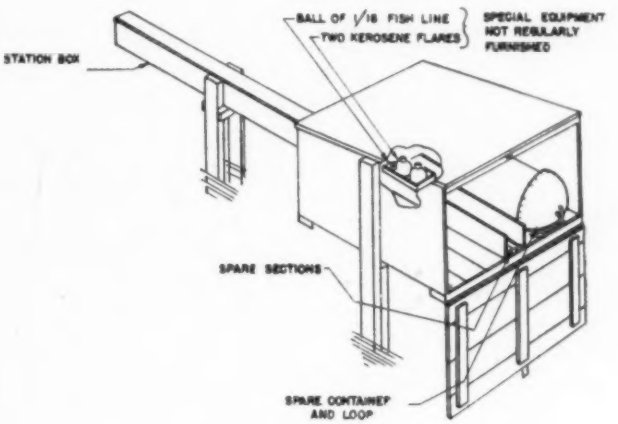
The container is made of flexible materials to withstand the rough treatment to which it is exposed. At the present time a composite construction of fiber, rubber, and canvas is doing a very fine job.

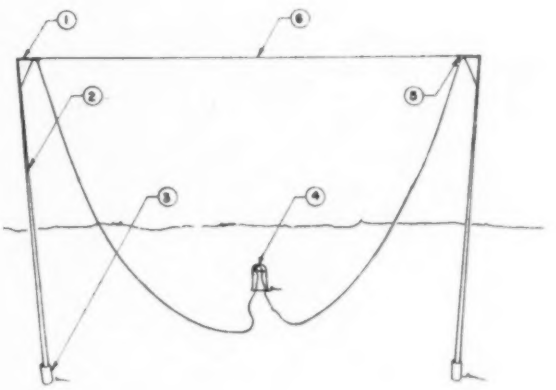
Concurrent with the improvements in the energy absorption units, or pickup units as they are called, was the simplification of the station and contact means. Pilots found it unnecessary and undesirable to make pickups at a 100-ft altitude, and a better means of positioning the hook was necessary to ensure its contacting the loop. The modern pickup station is only 12 ft high and 20 ft wide, is portable, and is composed simply of two light poles. Fig. 5 illustrates a typical air-mail pickup station and shows the station box which serves the dual purpose of station marker and stowage space. A nylon rope loop attaches to light spring clips at the top of each pole and is threaded through an eye on the top of the container which rests on the ground.

With the smaller station a better means of contacting the loop was needed. The dangling grapnel hook had many disadvantages. The present hook and contact arm were developed to give a positive means of positioning the hook and guiding the loop accurately into it. This arrangement permits the pilot to make last moment corrections for cross winds or rough air by yawing or rolling slightly to guide the hook properly through the station. The length of the contact arm is such that for small ships the hook is about 7 ft below the wheels or propeller tip, as shown in Fig. 6. On large ships this distance is usually increased to 9-10 ft.

The air pickup planes do not handle loose mail. The Post Office Department has designed a special small pouch for pickup work and mail to and from each post office is handled in these. Each is fitted with a lock which may be opened only by post office employees. Several of these will fit inside of the standard pickup container used. The pouches have tags on them labeled by the post office according to their destination. This enables the operator to sort the pouches swiftly in flight and load the container to be dropped at the next station with a minimum of trouble.

To make a pickup and delivery the operator loads the pouches and then trails the container by its pickup loop outside the airplane. This loop is fastened in a release which is operated by the pilot just before the airplane goes through the pickup station. As soon as the outgoing container is ready, the operator reels a few feet of line off the drum, places the hook on the track of the pickup arm and lowers the arm into position. The hook slides down the track into a flexible retainer at the end of the arm. He then checks the brake setting and notifies the pilot. When the pickup station comes into view the airplane goes into a power glide at about a 7:1 angle and at 130 mph. A hundred feet ahead of the station the mail delivery is made and a few feet from the station the airplane begins its climb. The pickup arm strikes the loop which slides down the arm and engages the hook, pulling it out of the retainer. The nylon rope becomes taut, the drum is accelerated, and the brake comes on, accelerating the load to airplane speed. It is then winched into the airplane. This has been repeated by actual performance as often as 32 times in 45 min, mail or express being removed from





■ Fig. 5 – Mail pickup station and station box

the container on each pickup. Fig. 7 illustrates the above operational procedure.

Just as the fast railroad trains carry mail from small communities without stopping by utilizing their hook mail pickup devices, so are 110 small communities in the United States serviced by the type of air-mail service described. The airlines of today should look forward to the future with an open mind, for it may not be long before airliners will have to emulate the railroads' ultra-modern mail service shown in Fig. 8. The economic success of air pickup is demonstrated by the fact that the Post Office Department has been able to realize a substantial profit on the air pickup mail service as it is being conducted today. Although the size and volume of express shipments so far have been limited, there is little doubt but that direct cargo pickup as well as mail pickup will become a common feature in the postwar world.

Air cargo and air-mail pickup may be carried on with or without passengers. To date, only special passengers have been occasionally carried on the pickup airline. It is

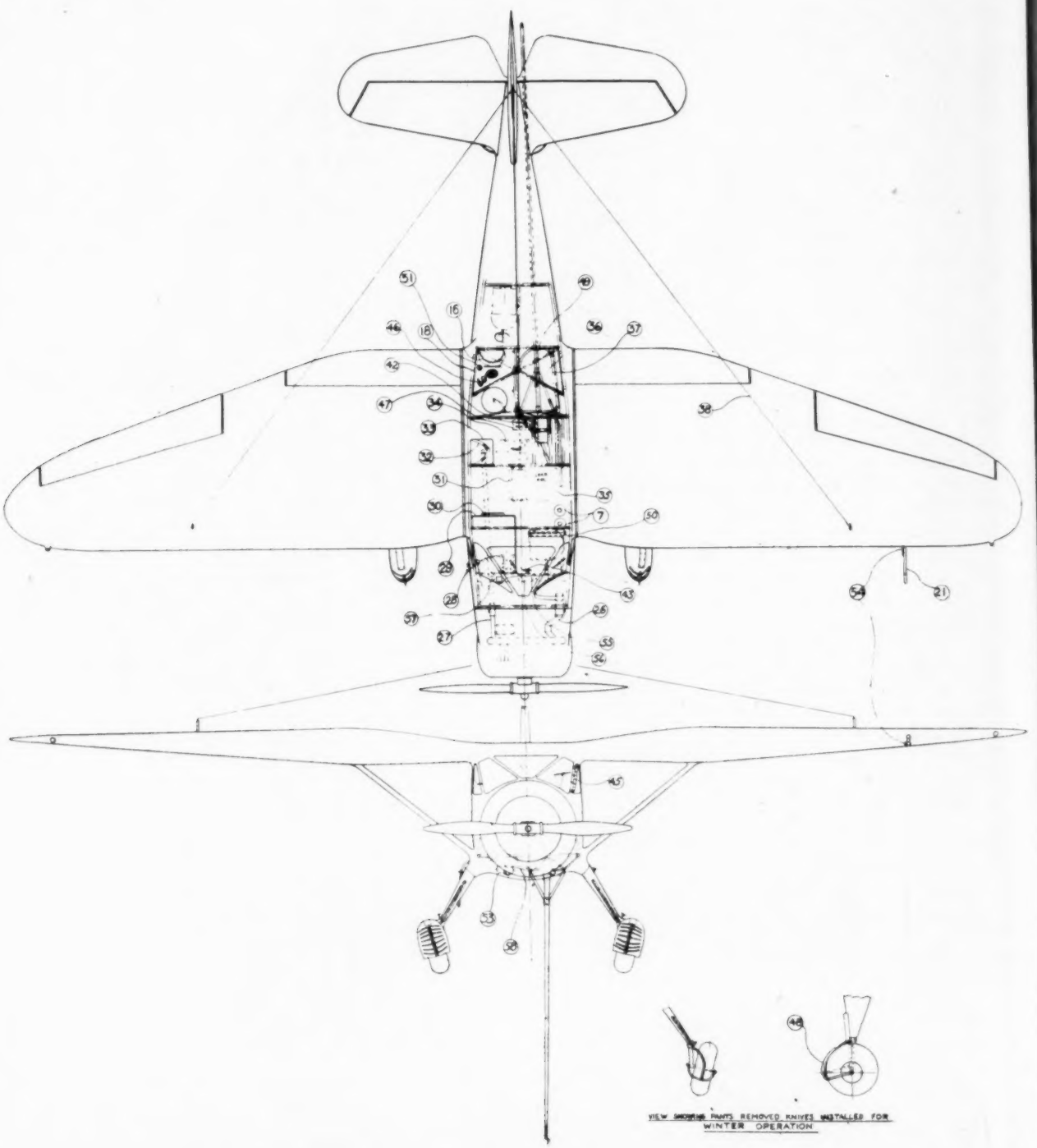


Fig. 6 – SR-10C airplane used by All American Aviation, Inc.

	4 oz	60	SAFETY SNAP
	4 oz	59	HOOK - TERMINAL BAG
		58	WIRING SYSTEM
	6 oz	57	SUCTION GAUGE
		56	DIRECTIONAL GYRO COMPASS
	6 oz	55	CLOCK
		54	OUTSIDE TEMPERATURE GAUGE
	3 lbs 8 oz	53	CONTROL BOX - 25 AMP
18 oz		52	BUTTON CONTROL - HEATER SYSTEM
	2"	51	EMERGENCY RATCHET & HANDLE - WIND IN
	5"	50	TOOL KIT - PICK-UP MAN
	7 oz	49	PARACHUTE DELIVERY BRACKET
—	—	48	KNIFE - WINTER USE
	5"	47	RESISTOR BOX
	5"	46	HOOK & LINE - EXTRA
8 oz		45	WIND DEFLECTOR - WINDSHIELD - PILOTS SIDE
	4 oz	44	SPONGE RUBBER BUMPER - STAB. STRUT
	7 oz	43	T-HANDLE - RELEASE
	2" 8 oz	42	RELEASE
	16"	41	REAR COMPARTMENT - STORAGE
	15"	40	CONTAINER - SPARE
	4"	39	ADAPTOR - WINCH BASE
	4"	38	AERIAL - RADIO - SENDING & RECEIVING
2"		37	HATCH COVER
—	—	36	HATCHWAY
	43" 10 oz	35	RECEIVER - LEAR
	12"	34	TRANSMITTER - LEAR
	4" 6 oz	33	PICK-UP MAN'S SEAT
	2"	32	SAFETY BELT - PICK-UP MAN
	38"	31	BATTERY 12 VOLT
	6 oz	30	LOG BOOK RACK
	11"	29	FRONT MAIL BIN - PHENOLITE
—	—	28	FUSE BOX
5"		27	HEATER - PICK-UP MAN
5"		26	HEATER - PILOT - WINDSHIELD DEFROSTER
	30"	25	GENERATOR 25 AMP - 15 VOLTS
—	—	24	OIL TANK - CONVERTED
	5 lbs 8 oz	23	WINDSHIELD WIPER
	4 oz	22	KEY & HOLDER - DOOR - STINSON
		21	PITOT - STATIC TUBE - HEATED
	1" 3 oz	20	TETHER BELT ASSEM - SAFETY - PICK-UP MAN
12 lbs		19	STEEL GUARD BARS
	3"	18	CANVAS MAIL BIN
	15"	17	CONTAINER
	3" 4 oz	16	TERMINAL MAIL BAG - CANVAS
	1"	15	SELF ALIGNING TAIL WHEEL
30"		14	METAL BELLY
	3"	13	LINE - PICK-UP
	13"	12	HOOK
38"		11	ARM & YOKE ASSEMBLY
	1" 4 oz	10	GUARD RAIL
	51"	9	PICK-UP UNIT
	8 oz	8	LOCKS 2
	1" 3 oz 2 oz	7	PARACHUTE DELIVERY CANS
	1" 5 oz	6	REINFORCING PAD - STEP - PANTS
	4" 4 oz	5	KNIFE - LANDING GEAR STRUT
	1" 3 oz	4	REINFORCING STRIPS - PANTS
	4" 3 oz	3	KNIFE - PANTS
	9 oz	2	EXHAUST STACK FAIRING - SPEED RING
EST'M'D ACTIVE		1	STINSON SR 10C CONVERSION - FULLY EQUIPPED FOR PICK-UP OPERATION

not known whether flights in rough air at low altitude with frequent pull-ups will be palatable to the public or not. More will be known about this after a year's operation of new passenger-pickup routes now in the proposal stage, but no serious adverse reaction is anticipated. In the discussion of the design requirements for pickup aircraft, it will be assumed that the combination service will be demanded by the flying public. The designs are discussed as intelligently as possible in view of the limited knowledge presently available.

The foregoing discussion has been largely concerned with elementary principles, the primary purpose being to acquaint the reader with the basic fundamentals involved so that he will have a rational outlook on the airplane design considerations which follow.

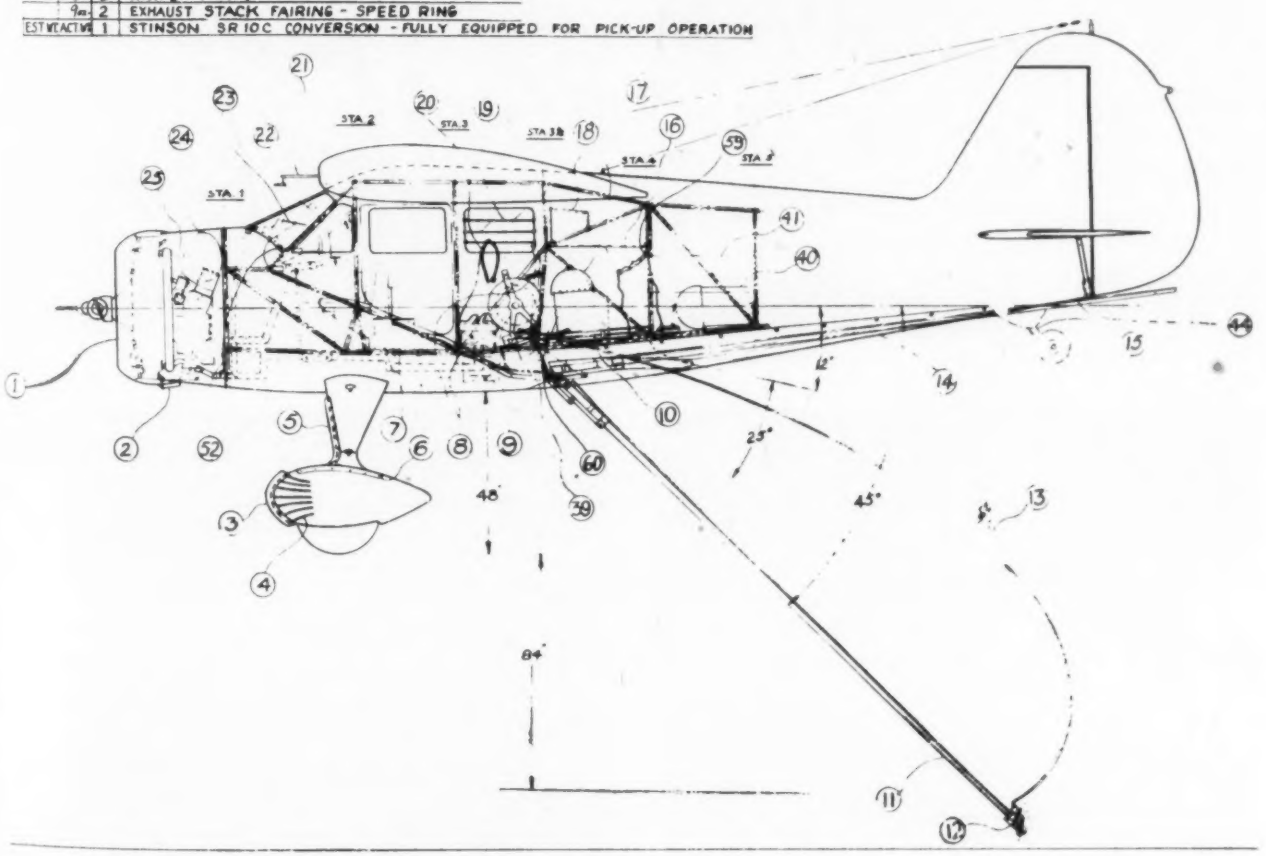
■ Basic Features of Pickup Planes

There are a few basic, desirable features that should be considered in the design of pickup airplanes. These include:

1. High wing design preferred: The high wing design allows the fuselage to be closer to the ground, reduces possibility of damage to the wing in the event that an approach to the pickup station is too low, and permits good visibility for the pilot down and to the side.

2. Twin-motored equipment preferred: Twin motors allow operations to be carried on at night and also permit transport of passengers according to CAA requirements.

3. Loading through a hatch in the belly preferred: Belly loading allows the load to be hoisted inside in flight more easily than through a side hatch.



4. Pickup operations to be carried on by copilot: This means that the pickup installation should be in the forward end of the airplane for convenience, for reduction of the number of personnel required, and to allow the operator to alternate as copilot during take-offs, landings, or while on instruments. Also, while on pickup duty, he will be able immediately to contact the pilot and exchange information with him.

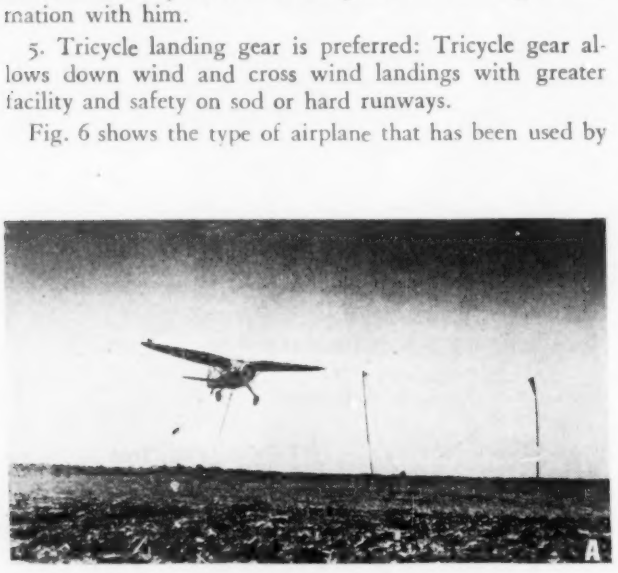
5. Tricycle landing gear is preferred: Tricycle gear allows down wind and cross wind landings with greater facility and safety on sod or hard runways.

Fig. 6 shows the type of airplane that has been used by All American for mail pickup for more than five years. Although the floor location of the unit eliminates the need for a rope guide pulley, it takes up valuable floor space and makes additional work for the operator.

Figs. 9 and 10 show proposed installations in two well-known airplanes which are available today. The overhead location of the unit is essential for the handling of heavier loads than now handled by the ship shown in Fig. 6.

■ Pickup Units

It is safe to say that a 200-lb capacity pickup unit is as large as will be used in cargo or mail pickup lines in the



■ Fig. 7 – ***A:*** Pickup plane approaching station with trailing bag about to be delivered; ***B:*** pickup plane delivering incoming mail container, just before contacting station to pick up outgoing mail; ***C:*** pickup plane at moment of contact with nylon rope fastened to container of outbound mail; ***D:*** pickup plane with hook just pulled loose from pickup arm and container about to be airborne; ***E:*** pickup plane has just left pickup station with outbound mail-delivered container with inbound mail may be seen in foreground while at extreme left is local messenger who carries mail between pickup station and local post office

United States for the next few years. This size (shown in Fig. 11) will be used in the 25,000-lb passenger-pickup airplane specified by the Feeder Airlines Association, specifications of which were made public in September, 1944, and which are included in this paper as Appendix I. It would also be used in the smaller 15,000-lb passenger-pickup airplane desired by All American for the smallest size plane believed feasible to operate economically. Specifications of this were made public in a talk the speaker gave before the Institute of Aeronautical Sciences at New York on Sept. 15, 1944, excerpts from which are included as Appendix II. The total installed weight of this pickup equipment will be about 230 lb.

The smallest type that we will consider in this paper is the 60-lb capacity unit shown in Fig. 7. This will be used in the smaller planes and is the unit which All American has been using in its Stinson Reliants for the past five years. Most of the mail pickup applications now on file contemplate use of this size.

Airplanes will vary from single-engine planes of 3500-lb gross weight up to the twin-engine Beech Model 18-S weighing 8000 lb. Although of smaller capacity it is also much lighter in weight, the complete installation weighing approximately 175 lb.

Although cargo pickup units having as much as a ton capacity are entirely feasible, our remarks here will be concerned with the pickup aircraft using these two smaller capacities of pickup units.

■ Design Requirements—200-lb Capacity

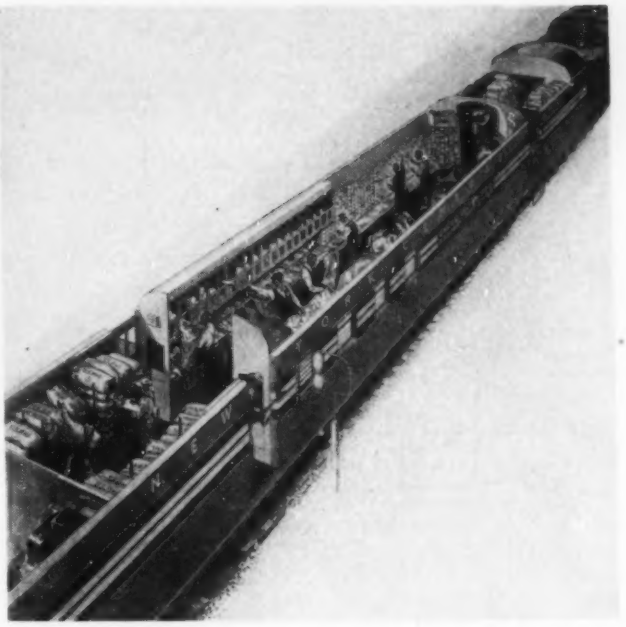
Let us assume that an airplane agreeing with the desired design features outlined in Appendixes I and II is to be used. The number of passengers carried may be varied to increase or decrease the cargo volume available in the airplane. The capacity of the pickup unit will be 200 lb.

Unit Mount – The unit is best mounted under the roof of the cabin so that it may winch the load entirely into the airplane. It may be advisable to mount it on rails so that the cargo may be stowed by using the unit as a traveling hoist with overhead rail suspension. Of course, it will be necessary to lock the unit in position for the pickups.

As for the structural design of the unit mount, the most conservative condition should be considered. In determining the loads applied it is assumed that the drum is jammed somehow or the brake delay is not properly set, that the pilot hooks an immovable object on the ground and that the line is new and over strength. With this exceedingly unlikely combination our factors would be as follows:

1. Breaking strength of the pickup line = 1.0.
2. 10% overstrength material factor for line = 1.1.
3. Limit load is then 1.1 times breaking strength of line, assuming 100% efficiency of rope splices.
4. Factor of safety to prevent permanent set = 1.5.
5. Design load = 1.65 times the breaking strength of the rope.

Although the maximum braking force of the unit which picks up 200-lb loads is only 1500 lb, and the nylon loops have sufficient elasticity to keep the drum accelerating force below this value, the breaking strength of the pickup line used tests 3800 lb. This apparent over strength is used in order to permit considerable wear before the rope must be discarded. Now, 3800 x 1.65 = 6280 lb, which is more than four times the 1500 lb normally to be expected and



■ Fig. 8 – Ultra-modern mail service of railroads may soon have to be emulated by airlines

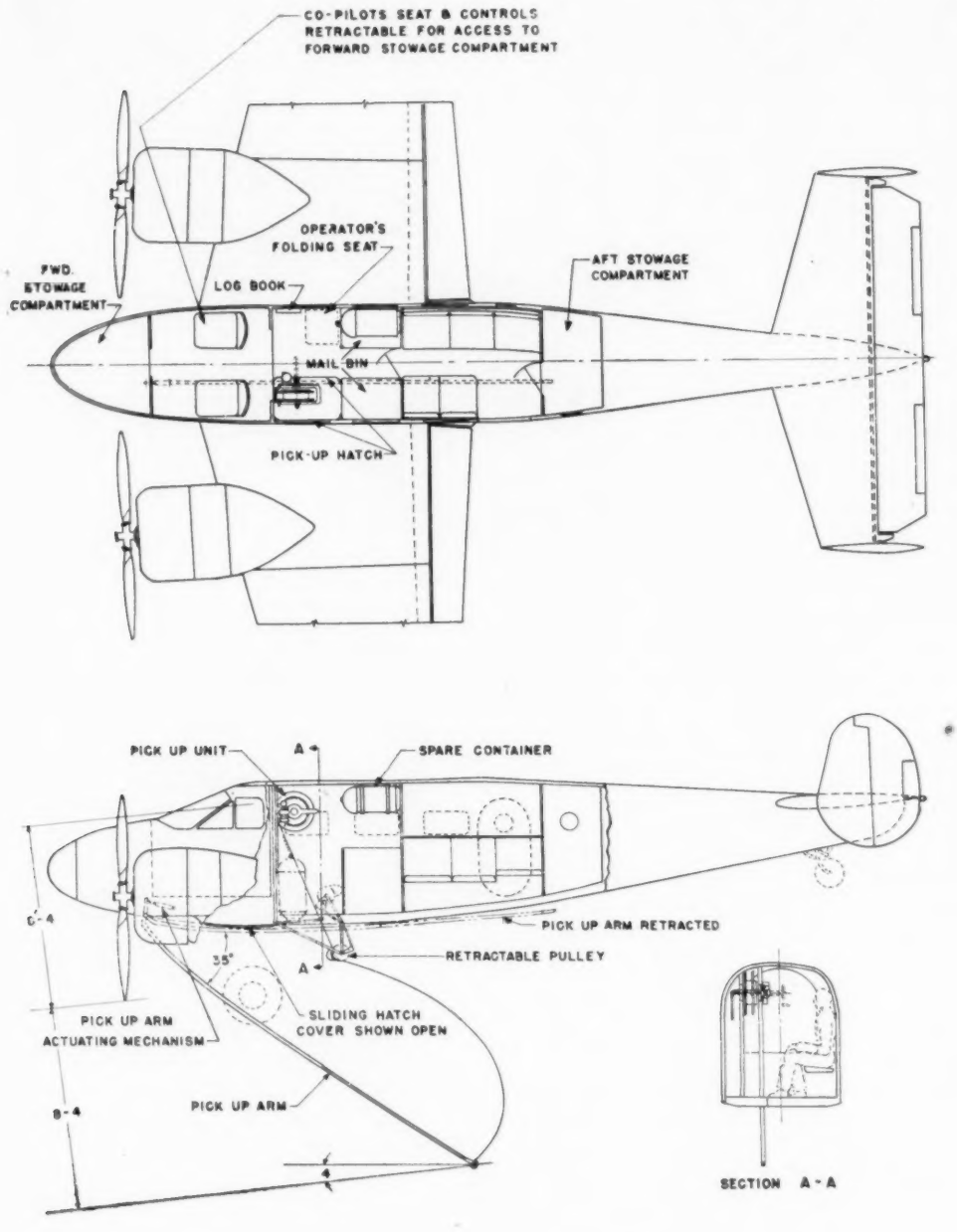
would result in an economically unsound structure. However, to design the mount to this unreasonable load would not take into account certain ameliorating factors, among which are:

1. The chances of the unit being jammed are extremely remote.
2. A trained unit operator might forget the brake delay setting once in 10,000 times.
3. The chances of the line developing full strength are probably 1 to 100 as it will almost surely be worn some and weakened where it wraps around the pickup drum or is bent around the pulley. The probability is that it will not develop over 2/3 its breaking strength.
4. The chances of hooking an object springy enough to allow the rope to develop the full strength of the mount before it is snapped by the inertia of the object and the unit is perhaps 1 in 10.
5. The probability that the cable is peeling off at a critical loading condition is about 1 in 5.

Therefore, with the fantastically small chance of ever having to deal with the 1.65 factor, it has been the practice to design to 1.25 times the nominal breaking strength of the line, or 4750 lb. This ensures that if the worst ever happens, the only damage will be a slight permanent set in the unit attachment. In those places where rigidity and ease of maintenance are desirable, however, higher margins of safety are advisable. This factor of 1.25, then, provides safety with a minimum of surplus structural weight.

It is customary to design the unit attachment for four loading conditions:

1. Line peeling off tangentially from left side of drum, simulating maximum braking force.
2. Line peeling off tangentially from right side of drum, simulating maximum braking force.
3. Line pulling from left side but at centerline of drum shaft, simulating drum accelerating force.
4. Line pulling from right side but at centerline of drum



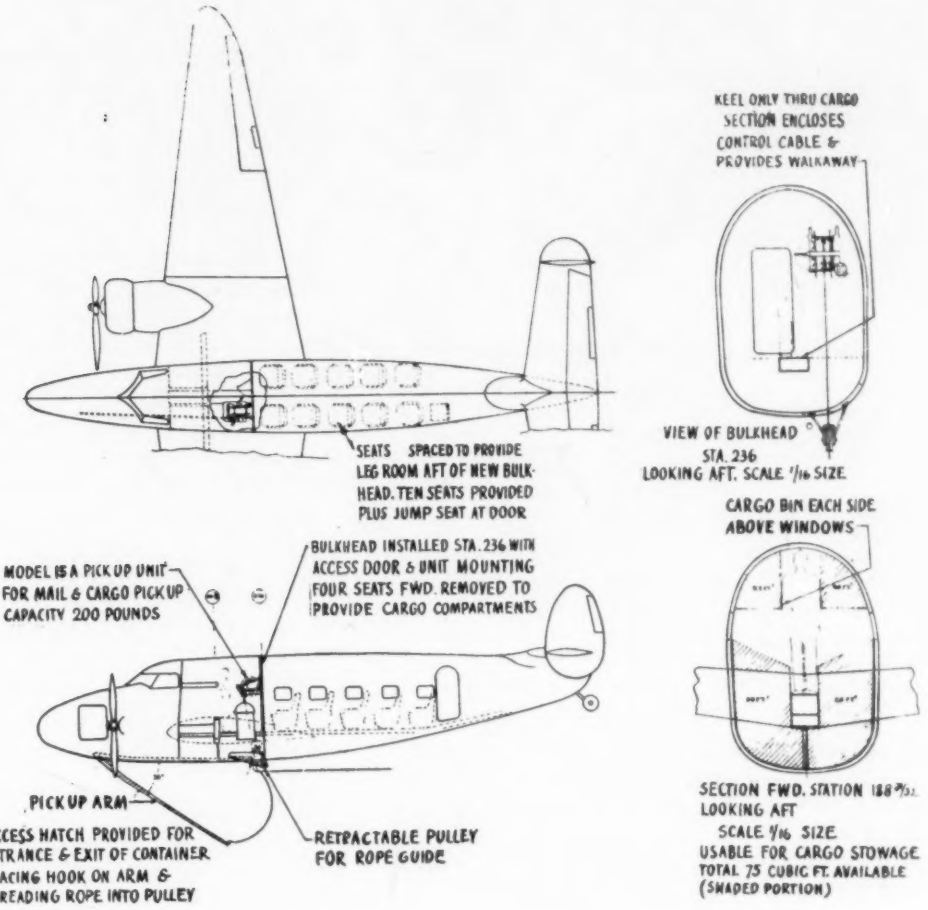
■ Fig. 9 – Pickup passenger arrangement proposed for twin-engine Beech 18-S weighing 8000 lb

shaft, simulating drum accelerating force.

Hatch – Below and perhaps a little aft of the pickup unit is the loading hatch. This is most usually located in the floor of the airplane. The cover for the hatch should close flush with the skin of the airplane, and be operable so that it does not present an opportunity for the cargo to catch on it, nor should it be jammed or closed by frozen mud or ice. Its dimensions should allow reasonably bulky cargo to be loaded with ease. A hatch 50 x 30 in. would be desirable, although physical limitations usually permit only a smaller hole, such as 25 x 20.

Guide Pulley – A means to guide the pickup line out of the airplane during pickups and to prevent light cargoes from beating against the fuselage is the next step in the installation. This usually consists of a pulley, generally retractable, so that the pickup may be made with the pulley close to the skin of the airplane to take this load directly into the airplane structure. The design load for the pulley and the close-in mounting should be based on the 1.25 factor. Before the cargo gets too close to the belly, the pulley may be extended. The design factor in the extended position may be much lower. The mechanical design of the pulley mounting is usually tricky and experience is required to match the proper pulley design to the particular airplane. Retractable pulley arrangements are indicated in Figs. 9 and 10 and shown in more detail in Fig. 12.

Contact Arm – The contact or pickup arm must be placed so that it hinges forward of the hatch. It is desirable to hinge it so far forward that it will be possible


KEEL ONLY THRU CARGO SECTION ENCLOSES CONTROL CABLE & PROVIDES WALKAWAY
VIEW OF BULKHEAD STA. 236 LOOKING AFT. SCALE 1/16 SIZE
SEATS SPACED TO PROVIDE LEG ROOM AFT OF NEW BULKHEAD. TEN SEATS PROVIDED PLUS JUMP SEAT AT DOOR
CARGO BIN EACH SIDE ABOVE WINDOWS
MODEL IS A PICK UP UNIT FOR MAIL & CARGO PICKUP CAPACITY 200 POUNDS
BULKHEAD INSTALLED STA. 236 WITH ACCESS DOOR & UNIT MOUNTING FOUR SEATS FWD. REMOVED TO PROVIDE CARGO COMPARTMENTS
PICK UP ARM
ACCESS HATCH PROVIDED FOR ENTRANCE & EXIT OF CONTAINER. PLACING HOOK ON ARM & THREADING ROPE INTO PULLEY
RETRACTABLE PULLEY FOR ROPE GUIDE
SECTION FWD. STATION 188 3/32 LOOKING AFT SCALE 1/16 SIZE
USABLE FOR CARGO STOWAGE TOTAL 75 CUBIC FT. AVAILABLE (SHADED PORTION)

▪ Fig. 10 – Pickup passenger arrangement proposed for Lockheed 18

to place the hook directly into the hook retainer. If that is not possible it should hinge as far forward as good design permits so that the track which guides the pickup hook into the hook retainer will be as short as possible and so that the extended arm will not interfere with pulling in heavy containers which hang nearly straight down. The hook should be loaded onto the arm through the pickup hatch without requiring the operator to extend more than his hand into the wind. The arm should be lowered to about 35 deg with the fuselage reference line, preferably by a mechanical actuator. Typical contact arm arrangements are shown in Figs. 13, 14, and 15.

For various practical reasons the lower portion of the arm is usually made of wood. The size of the wooden portion has been determined largely by experience and is usually 1½ in. in diameter at the tip and 2 in. in diameter where it slips into the steel portion. The wood portion should be at least 12 ft long and made of oak, hickory, white ash, or similar material.

The load used to design the yoke of the pickup arm is determined by the force required to break the wooden portion of the pickup arm applied to the tip at right angles to a radius from the pivot point, times the strength factor of 1.5. Reasonable margins of safety should be shown to ensure the proper stiffness of the yoke. Margins of 50% are usually desired to prevent fatigue cracks caused by quivering of the arm after contacting loops.

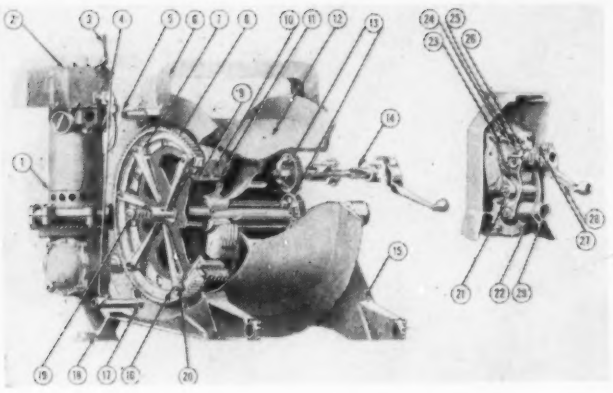
■ Design Requirements – 60-lb Capacity

Unit Mount – The unit should be mounted, if possible, on the ceiling as recommended for the 200-lb capacity model. The same design considerations are used but with a 1700-lb breaking strength nylon rope; the 1.25 factor giving a 2125-lb design load.

The same four loading conditions as outlined for the 200-lb capacity unit are used.

Hatch – The hatch for this unit is similar to that recommended for the larger unit, except that it may be as small as 20 by 25 in., with rounded corners of about 6-in. radius allowable.

Guide Pulley – Fig. 6 shows an installation which has been used for years on Stinson SR-10C airplanes. It is to be noted that there is no pulley. This has the drawback that rope wear tends to be excessive, and a reinforced belly is required to withstand the banging of empty containers. It is recommended therefore, that a pulley installation be carefully considered; the design factor on this pulley to be 1.25 times the breaking strength of the rope in pickup position, and lower for winching operations, as outlined earlier.



■ Fig. 11 – Sectional assembly view of 200-lb pickup unit

1. Motor
2. Electrical controls
3. Pawl control
4. Pawl control rod
5. Hand wind
6. Tie bar
7. Clevis key
8. Ring gear
9. Brake support
10. Brake discs
11. Brake pressure plate
12. Drum
13. Countershaft gearing
14. Brake operating nut
15. Time adjustment support
16. Setscrew
17. Pawl
18. Motor support
19. Brake application tube
20. Brake lever
21. Brake operating yoke
22. Brake adjustment barrel
23. Time adjustment lever
24. Time adjustment knob
25. Time adjustment stop
26. Time adjustment cam
27. Brake resetting crank key
28. Brake resetting crank
29. Brake adjustment knob

Contact Arm – Since the size and weight of the pickup loop on the container designs the contact arm, and since the loops on both models of unit are similar, it has been found desirable to use the same arm on installation of either size unit. A slightly smaller arm, 15 ft long, 1⅜ in. in diameter at the small end and 1¾ in. at the upper end has been used on the Reliants. It was found necessary, however, to tape and dope the upper end for 5 ft. to prevent excessive breakage.

■ Mail and Cargo Bins

The principle design features of mail and cargo bins are, in order:

1. Safety.
2. Lightness.
3. Accessibility and capacity.
4. Seeability.

It is wise, perhaps, to design the mail and cargo bins for densities of 25 lb per cu ft or more and with suitable structure forward for protection of personnel in a crash condition. Other accelerated flight conditions must be considered according to the requirements of the CAA.

Wherever possible, the bins should be made accessible to the operator and within easy reach. Doors on the bins might be replaced with elastic nets. The capacity might be increased in several ways: one of these is to have "balcony" bins in odd locations, another is to utilize the space behind the pilot's seat (for example) by having the forward side of the lower part of the bin made of a strong net which allows cargo to be loaded to fit to any location of the adjustable seat.

Seeability will prevent errors which might occur if a mail bag or express package were to be left in a dark corner of an opaque cargo bin. Use of screening for bin floors, or transparent plastic, will be a great aid in this respect. Lighting, either natural (through windows) or electric, is quite necessary.

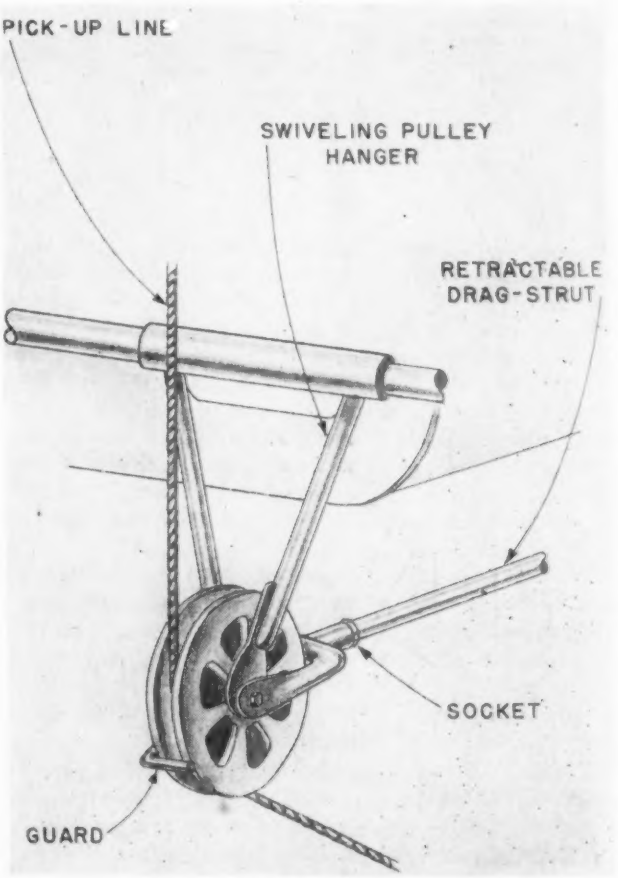
■ Miscellaneous Requirements

The winch action of the pickup unit is powered by a 12-v or 24-v motor. The capacity of the electrical systems must be sufficient to power the motor without interfering with the normal electrical requirements of the airplane. The 60-lb unit has a ¾-hp motor, and the 200-lb unit a 1¾-hp motor.

Heating the airplane is no small task if one considers the fact that a large hatch is opened periodically. It is recommended that heat sources be available to the hands and feet of both the pilot and operator as well as to raise the temperature of the cabin to a reasonable point. This indicates a heating unit which could give perhaps twice the Btu's required if no hatch were opened. No carbon monoxide should be present either from the heater or from the engine through the open hatch.

A good windshield wiper is a requirement that cannot be overlooked. It must always be kept in mind that excellent pilot vision is very necessary at all times at low altitudes and at 120 mph or faster. A glass window, which can be opened for direct forward visibility, is also customary.

Radio reception is of prime importance and every effort must be made to give the best possible transmission and



■ Fig. 12 – Retractable guide pulley

reception between the airplane and the terminal and way stations. Distances up to 350 miles with mountains separating plane and terminal should be assumed.

The foregoing remarks are deliberately conservative in scope of unit capacity. They were predicated upon immediate requirements of pickup. However, at the present time there are pickup units available that could be adapted to snatch a ton from the mine, jungle, boat, or factory roof with ease. These and larger units will be available in the near future. At the present time we have no airplanes designed to make such pickups although there are several airplanes of the proper weight and power that might be converted to accomplish this mission.

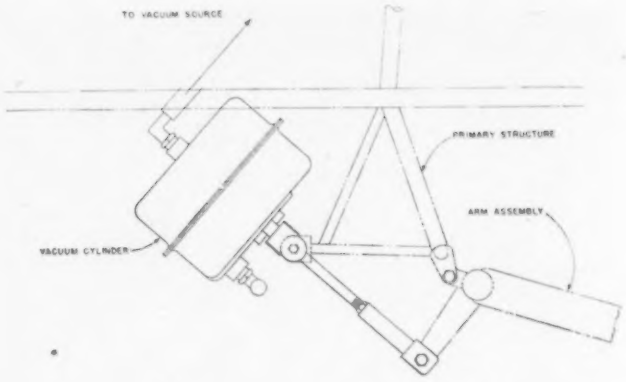
■ Glider Pickup

So far, we have dealt only with the direct pickup of loads. If we desire to pick up gliders several advantages are achieved: First, the same accelerating force that would pick up 1000 lb directly at 8g will pick up an 8000-lb glider at an average of 1g. This glider will carry a payload of 4000 lb on an average, and if well designed, nearly 5000 lb. With the tow line extending between the glider and tug, transfer of cargo between the two would not be insurmountable. Blind towing and towing in turbulent air is no longer a problem, thus the future of the often discussed glider train is not blocked by matters of technique. It is not the purpose of this paper to go into the why's and wherefore's of glider towing except to point out that for short haul and for certain special operations, air transport via glider pickup has excellent possibilities of usefulness.

We again are indebted to the genius of the late Richard C. duPont for developing this phase of pickup work. The immediate benefactors of this idea have been the U. S. Army Air Forces. Conversions of many types of military aircraft for picking up gliders have been made in the past two years or so, the best known of which is the C-47 – CG4A combination which has proved itself on Burmese and Normandy battle fields as well as at many training bases for glider borne troops. Thousands of successful pickups have been made under good and bad conditions. The military usefulness of this combination is well illustrated by the sequence shown on Fig. 16.

■ Glider Pickup Units

Glider pickup units differ from cargo pickup primarily in the amount of cable on the drum and the thermal capacity of the braking elements. It has been found that

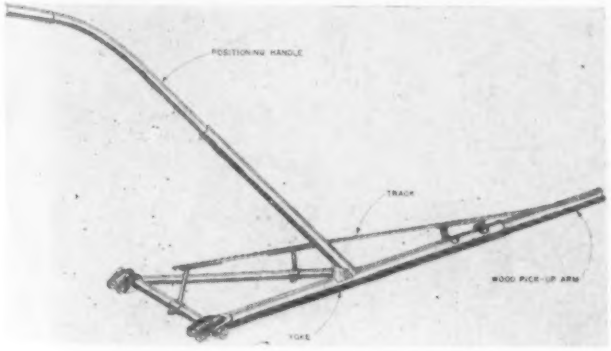


■ Fig. 14 – Vacuum operated arm used on SR-10C airplane

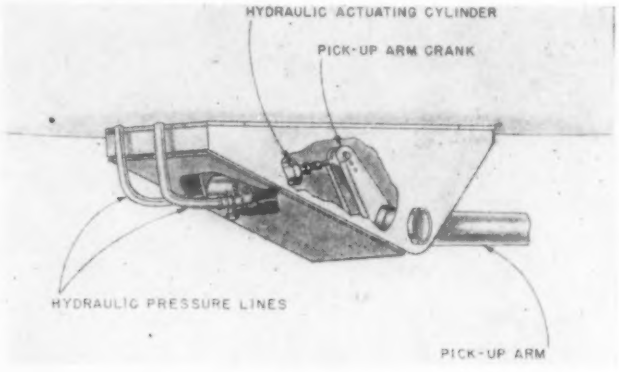
0.7 to 1.0g is the practical range for glider acceleration, due to various considerations such as pickup forces on the tow plug attachment and effect on the glider passengers. A 0.7g pickup at 130 mph average speed requires a payout of 800 ft of cable, so, allowing a reasonable amount extra for safety, the cable length usually exceeds 1000 ft. Curves of duration and payout are shown in Figs. 17 and 18. These are similar to the curves, Figs. 2 and 3, previously shown for cargo pickup.

Cable strength is controlled by the 1.0g pickup. It has been found through experience that surges will occur during the pickup to the extent of 1.3g. Inasmuch as it is not desirable to exceed the yield point of the steel cable it is customary to use a cable with a breaking strength nearly twice as strong ($1.5 \times 1.3 = 1.95$) as the normal (1.0g) capacity of the unit.

The thermal capacity of the brake is a very critical factor. Fig. 19 shows the kinetic energy necessary to be absorbed by the brake to accelerate various weights of gliders to various speeds. The braking elements reach rather high temperatures and cooling air ducts are provided for rapid cooling in cases where series pickups or rapid training flights are to be made. Overloading the units either by excessive speed, overweight, or too little cooling in between pickups may result in rapid deterioration of the braking elements. Due to the large contribution the brakes make to the total weight of the unit, these are designed to much smaller margins of overload than on cargo pickup units.

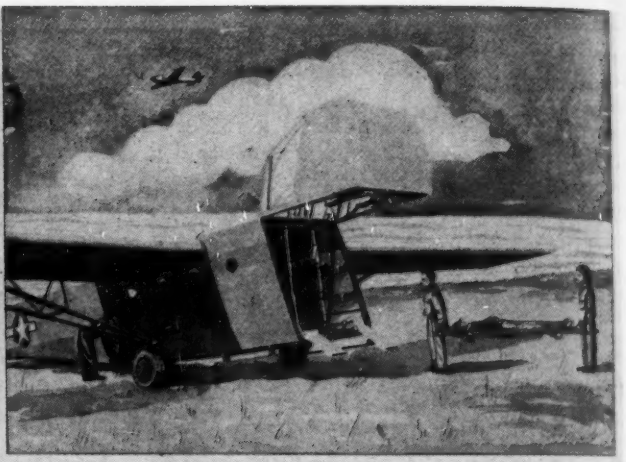


■ Fig. 13 – Manually actuated contact arm

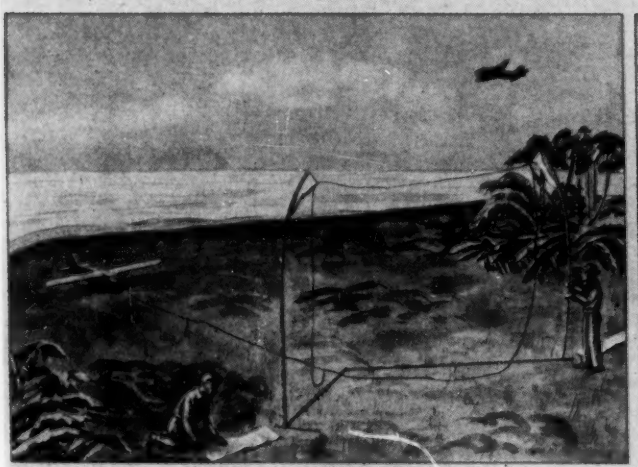


■ Fig. 15 – Hydraulically actuated arm

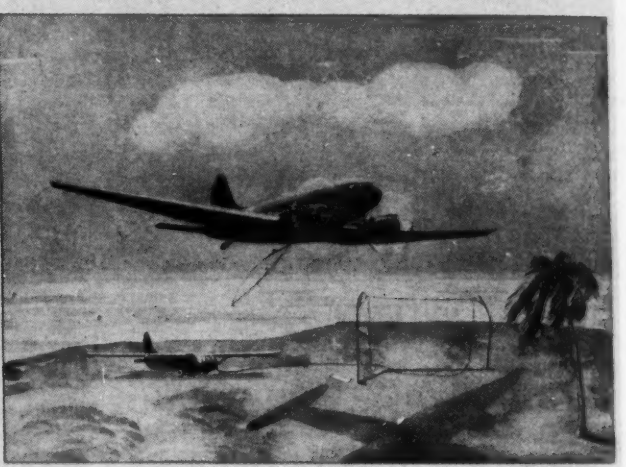
GLIDER RELIEF LANDING



RAISING NOSE AND LOADING INJURED



ERECTING PICK UP STATION



PICK UP RUN

EVACUATION

■ Fig. 16 – AAF glider evacuation unit – thousands of successful pickups have been made under good and bad conditions – military usefulness of glider pickup is obvious

It is necessary to lay the cable on the drum smoothly, so level wind devices are standard equipment. It has been found that as many as six layers of cable will operate satisfactorily when properly wound on the drum. Larger and continuous duty rewind motors are required to haul the gliders up toward the airplane so they can be transferred from the tow cable to the normal tow fitting. Substantial guards are necessary to enclose the drum and other mechanism. A typical glider pickup unit is illustrated in Fig. 20, which shows the Model 80 pickup unit used in the U. S. Army Air Forces' C-47 airplanes for picking up CG-4A gliders.

The weight of glider pickup units and their mountings and adjacent auxiliary equipment runs about 10% of the weight of the glider they are designed to pick up. Other auxiliary equipment, such as arm, hawse pipe, and guide pulleys, add another 2%. The effect of these weights on the c.g. of the airplane must be given careful consideration.

■ Design of Glider Pickup Airplanes

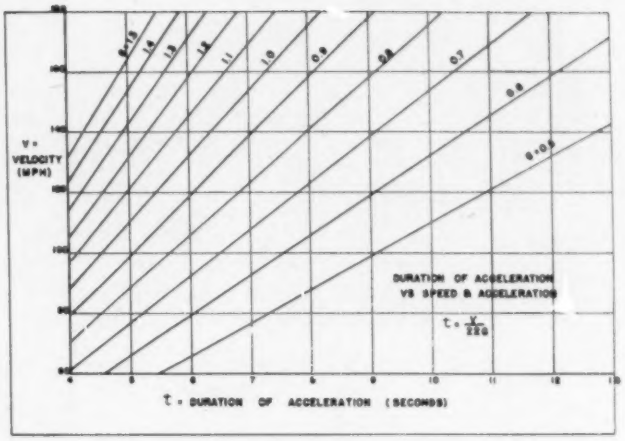
Experience and calculations indicate that an airplane should weigh at least twice as much as the glider to be picked up in order that the deceleration of the airplane be kept within reason. This will vary somewhat, particularly with the horsepower available, but it is a good rule of thumb to work with. Here again the high wing type shows some advantages, particularly with regard to vision as well as the fact that there is less chance that the wings will be damaged in a low approach.

Due to the magnitude of the pickup loads and the effect of the pickup unit installation on the center of gravity of the airplane, the unit is usually anchored to the region of the wing spars which is usually very rugged.

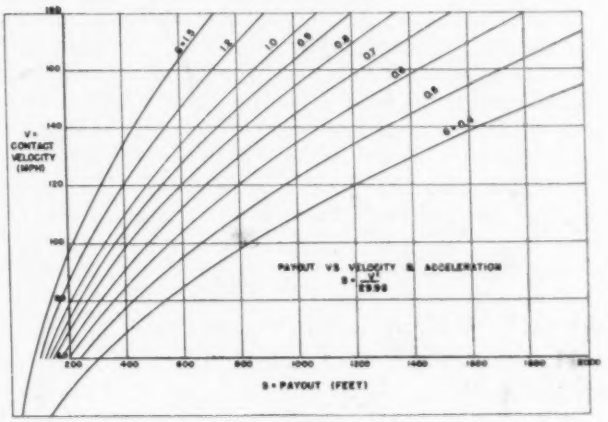
The pickup line can be guided aft and under the floor via pulleys through a long hawse pipe and can emerge from the fuselage at or near the tail. It is this point that makes the bomber-type fuselage so suitable, since the tail gunner's position allows the line to be serviced from the very tail end. The prime advantage here is the fact that the pickup line emerges aft of the tail surfaces and eliminates any elevator sawing tendencies that a high riding glider might induce. The point that so much leverage is allowed by the tail exit of the line does not bother the pilot at all, inasmuch as the tail position is the normal towing point for all aircraft. The controllability of the airplane is not adversely affected when the line emerges near the tail as this is very nearly in line with the c.g.

The aft position of the tow guide mechanism allows an easier solution to the possibility of the cargo-transfer-in-flight problem, as a transfer container could be launched and received through what in wartime is a tail gunner's position. Please do not interpret these examples: "bomber type fuselage" and "tail gunner's position" to mean that conversion of bombers is advocated. This is possible, but probably is economically unsound. Instead, these terms are used because there are no civilian names for similar commercial designs.

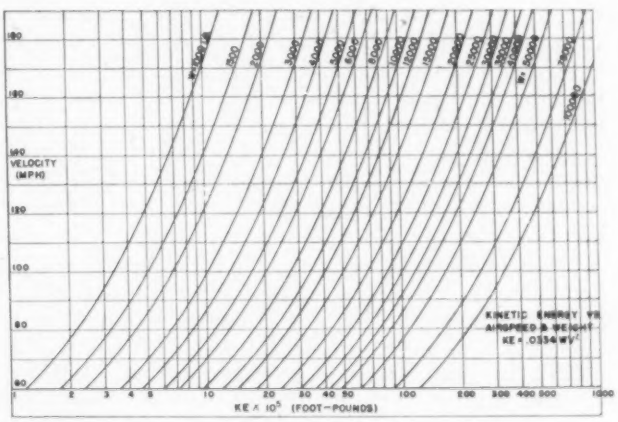
The technique of "snatching" gliders is not too unlike that described for pickup of mail and cargo. The units are bigger, but the method of preparing the airplane for pickup is otherwise the same. The one prime difference, however, is the pickup line, which is usually nylon rope in cargo work and steel cable in glider work. Hence, the elastic effect of nylon for accelerating a heavy drum of steel cable must be supplied in the loop and leader from the station to the glider. This leader is 200 to 450 ft long, depending upon the speed, weight, and so on, of the pickup combination. The length of the leader allows the glider to be set to the left of the pickup station and somewhat behind it in such a manner that the pickup airplane will make its approach to the right side of the glider. The



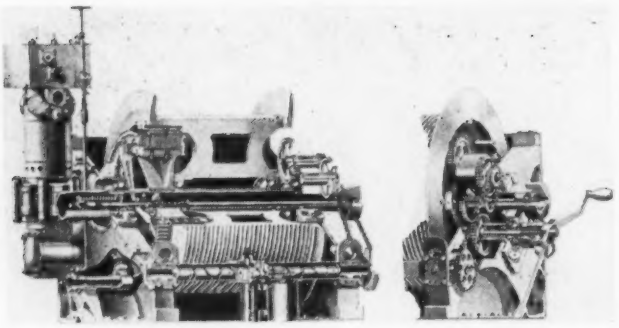
■ Fig. 17 – Duration of acceleration plotted against airplane speed and rate of acceleration



■ Fig. 18 – Payout of line plotted against airplane speed and rate of acceleration



■ Fig. 19 – Kinetic energy plotted against airplane speed and cargo weight



■ Fig. 20 – Sectional view of 8000-lb glider pickup unit

glider is off the ground before it reaches the pickup poles, but these are rigged so that they will fall down before the glider gets to them. See Fig. 21 for illustration of flight pattern.

Fig. 22 is useful in calculating the deceleration of the tug during a pickup. It is to be noted that this chart does not allow for the thrust of the propellers during the pickup cycle but is based entirely on kinetic energy transfer. Under average conditions of throttle advance the deceleration will be only three-fourths of that shown.

Also, the series pickup has been developed. This operation merely involves winching in the first glider and the transfer of its towline to a standard tow release. The pickup unit can then be reset and the tug, with one glider in tow, picks up another. This can be repeated as many times as practical, and has obvious commercial advantages.

■ Design Requirements

For the determination of the design loads to be used in the installation of glider pickup equipment, reference is made to Fig. 23, which is a typical trace made by an accelerometer or g-meter, which was firmly anchored to the floor of the glider being picked up. We are able to deduce from this typical pickup, points of interest to the designer of pickup airplanes and gliders. The chart is read from left to right. Each vertical line represents 1 sec, and the horizontal lines are increments equal to 0.1g or 3.2 ft per sec per sec. Multiplying the gross weight of the glider which was picked up by the g's developed on the accelerometer trace gives the approximate force in the tow line at any instant.

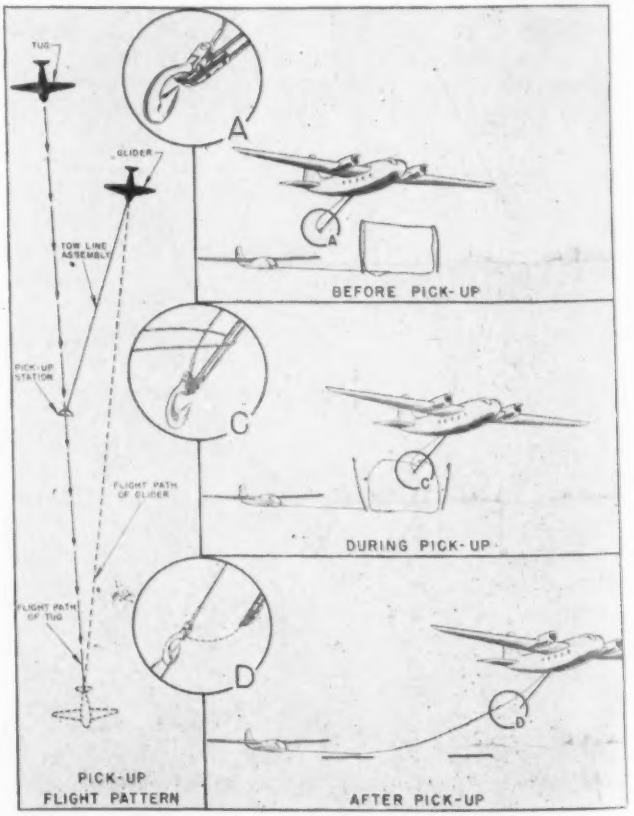
To be precise, an allowance of approximately 10% additional should be made for rolling friction at the beginning portion and air drag at the end portion. Indeed, when the glider is picked up from soft sand or with the brakes locked, the g-meter trace is quite meaningless. Likewise when the attitude of the glider changes from anything but horizontal, proper correction should be made for the influence of gravity. In test work, however, using concrete runways and skilled pilots these variables are well controlled. A recording tensiometer would be better than a recording accelerometer but as yet no satisfactory instrument has been available and the g-meter traces give sufficiently accurate data for design purposes.

It will be seen that about ¼ sec is necessary to accelerate the drum, at which point a peak of approximately 0.7g is recorded.

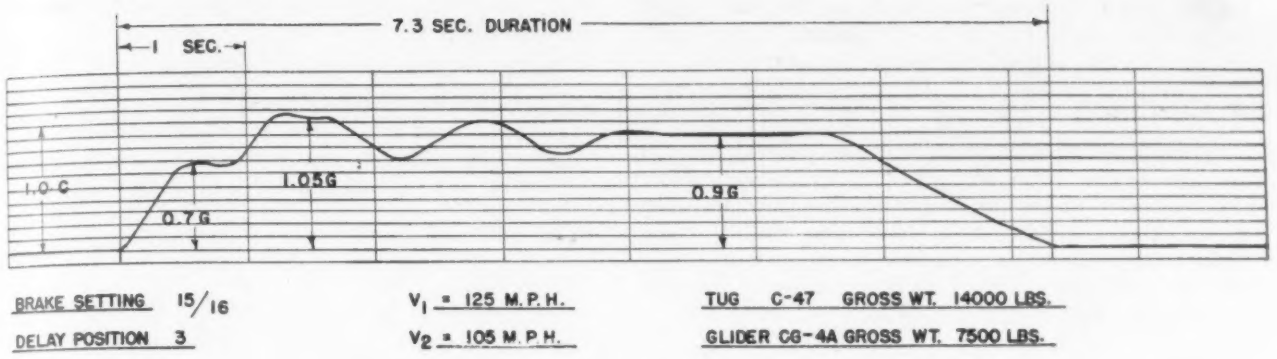
At this point the brake begins to come on and a little dip occurs in the trace before the brake comes full on. The braking force continues fairly uniform at 0.9g until the drum rotation stops, after which a tapering-off acceleration occurs due to the recoil of the nylon. Normally the braking force is not constant, there being a 15% fluctuation of the accelerating force due to various factors, primarily the action of the nylon behaving as a tension spring of about one-second frequency. Maximum surges of 1.2g occur occasionally, however, and this allowing 10% correction for unrecorded drag, establishes the limit load criterion of 1.30g limit load for proof test of the installation.

We may determine the ultimate loads applied to the airplane by the same general line of reasoning as used in direct pickup. With the larger units, the loads become very large. One of the medium sized units developed, having a capacity of 8000 lb, uses a line of 14,400 lb minimum breaking strength. This is multiplied by 1.25 as a safety factor. This gives $1.25 \times 14{,}400 = 18{,}000$ lb design load. The 1.25 factor is ample margin as the unit, cable, and mountings weigh in the neighborhood of 800 lb, and the inertia is such that cable breakage is probable before yielding of the airplane structure would occur. Likewise, as in direct pickup, the probability that all critical loading factors will occur at one time is extremely remote.

Unit Mount – Each installation is tested to a limit load equal to the capacity of the unit × 1.3, assuming the conservative condition of the load being applied while the drum is being accelerated as well as while accelerating the glider. The loads are applied to the unit and mount in four conditions in the same manner as for cargo pickup



■ Fig. 21 – Glider pickup flight pattern



■ Fig. 22 – Glider acceleration curve – Model 80 pickup unit

Tow Guide – Naturally, all of the pulley and hawse pipe connections to the fuselage must be stressed for sidewise components incident to cross wind pickups or temporary displacement of the glider during its ground run. Further, the fuselage itself must be strong enough to take these loads. In the same manner, vertical displacements of the load must be considered. Obviously, the sidewise and vertical combined angles of the towline are a stress possibility. The completed airplane is proof loaded to the 10,400-lb limit load outlined above. The forward reaction must be carefully applied to the landing gear and propeller hubs lest hidden damage to the airplane result.

Contact Arm – The arm should be hinged in such a way that little or no track need be used to put the hook in the hook retainer. Also, the arm should be in easy reach of the operator for this operation. Larger and longer arms are needed as the size of the airplanes, gliders, and units increases. The arm increases in diameter for stiffness as heavier nylon loops are employed. It is very impressive to view slow motion movies of the arm and to see how a 2½ in. diameter hickory pole about 12 ft long whips around as if it were rubber hose when it contacts a loop at 140 mph. Design of the arm mounting is on the same line of thought followed in cargo installations.

Access Hatch – Access to the pickup hook is required to prepare for the pickup. This hatch is located aft of the last tow guide pulley and is usually about 14 x 22 in. The door may swing up and inboard in such a way that the operator will not be impeded in his work. If the hatch is well aft a windshield may not be necessary, but if it is amidship, a small deflector is usually required.

Normally, this hatch is located in a section of the airplane which is subjected to both bending and torsion from the tail surfaces. It is, therefore, quite necessary to reinforce this cutout with suitable framing to assure that the strength will not be impaired.

EXAMPLE
Given: 1. Weight of C47=26,000 Lb
2. Weight of CG4A=8,000 Lb
3. Resultant Tow Speed Desired: 120 Mph

To Find: Required Contact Speed of C47

① Resultant Tow Speed Desired = 120 Mph

② KE of CG4A = 3.85

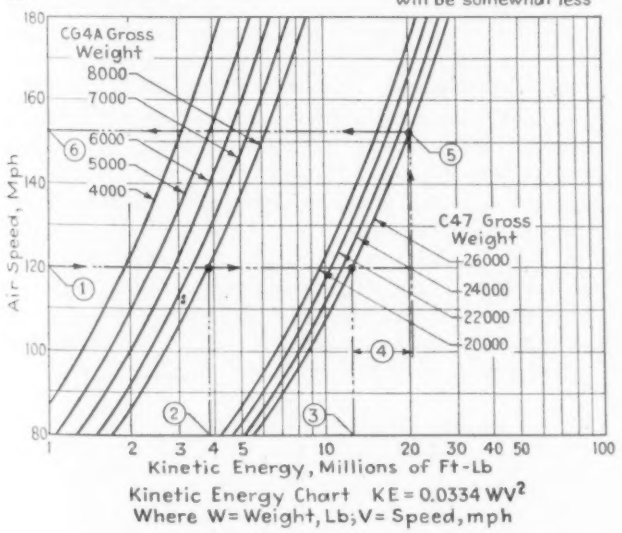
③ KE of C47=12.5

④ Multiply KE of CG4A by 2=7.7

⑤ Add 7.7 to KE of C47=20.2

⑥ Where Vertical Line at ⑤ Intersects C47-26000 Lb Line, Read Contact Air Speed Required at Left,= 153 Mph

Note: Actual reduction of Speed with normal C47 throttle advance will be somewhat less



■ Fig. 23 – Chart for determination of tug deceleration – C47 airplane and CG4A glider

■ Future Possibilities of Glider Pickup

Militarily speaking, glider pickup in warfare has tremendous value. It lends itself particularly well to those daring, tense moments where, by heroic measures, a battle may be won if fuel, equipment, or reinforcements are supplied in sufficient quantity, and especially in time. In Burma, in order to ferry a group of soldiers across a river too deep to ford, rubber boats equipped with outboard motors were landed in gliders on a sand bar. The job at hand was done and the gliders "snatched" the next day and returned to their base for re-use. Wounded and prisoners have been brought back, hospital gliders have been moved from place to place, and more tactical uses are being tested every day.

Its peacetime rescue value is obvious; explorers in wild countries and ordinary citizens in flooded areas will thank pickup for their lives.

In the realm of commerce we will see pickup stations

near factories, farms, mines, in jungles, on islands, and remote points of land; every place, in fact, that might want to tie into the many advantages and opportunities offered by a fast, flexible, low cost air transport system.

■ Conclusion

Sir George Cayley, in the Nineteenth Century, pointed out that the air is an ocean that comes to every man's door, and that we should take full advantage of this fact. Pickup of cargo and cargo gliders by aircraft in high-speed flight brings this prophecy to its ultimate realization. It provides a new tool with which the air transport industry can greatly widen the avenues of its future progress.

■ Acknowledgment

The author wishes to acknowledge the cooperation extended by the Army Air Forces in releasing much of this material and several of the illustrations for publication. Without their interest and support the rapid progress which has been made in air pickup during the past three years would not have been possible.

APPENDIX I

General Specifications for Local Service Airliners as Required by the Feeder Airlines Association

1. Ability to land and to take off in 1000 ft, fully loaded.

2. Movable bulkhead between forward cargo compartment and the seating section of the aircraft in such a manner that bulkhead can be moved and seats removed in a period of time not to exceed 10 min for the complete change. Last 12 seats to be permanent.

3. Seating capacity: minimum 18, maximum 22 (200 lb per passenger including baggage).

4. Maximum glide angle of 7 to 1.

5. Baggage space arranged so that each passenger can place his baggage when he gets in the ship and pick it up when he gets out, without any checking system.

6. Toilet (chemical), including washstand with running water. Drinking water in the ship.

7. Automatic device so that as the passenger door opens steps automatically let down into place without any outside assistance.

8. We prefer that the seating arrangement be such that seats will not need to be assigned due to load distribution.

9. The airplane designed structurally so that the pickup unit can be installed any time without structural change (designed to pick up a maximum of 300 lb using an installation similar to that used by All American). Complete pickup unit not to exceed 100 lb.

10. Tricycle landing gear designed to take care of 20 mph cross wind landings.

11. Retractable landing gear.

12. High wing.

13. Interior appearance and appointments to equal those of DC-3.

14. Manufacturer must keep in mind that all flying will be done at 500 to 1000 ft above surrounding terrain.

15. Landings will be made on an average of once every 50 miles, and at least 50% of landings will be made on turf fields; this calls for excellent brakes and oversize tires.

16. Two thousand pounds cargo payload (including the pickup gear).

17. Cruising speed at sea level 170 mph.

18. Range 500 miles with usual reserve.

19. Under-tank refueling with automatic cut-off.

20. All-metal airplane.

21. Size of passenger door, 30 in. wide and 70 in. high. Front cargo door 60 in. high and 60 in. wide. Rear cargo door 30 in. wide and 30 in. high. All doors to be on the left side.

22. If possible, provide a minimum of eight bins at least 18 in. by 24 in. and as deep as possible, with individual outside self-locking doors (to provide for small express packages and mail packages, these openings to be on the left side).

23. Individual reading lights and ventilators, with copilot call button arrangement for each passenger seat.

24. Ship to be air conditioned (covering heating and cooling).

25. Complete instrument and night-flying equipment, with full feathering propellers.

APPENDIX II

(Excerpts from "Passenger Airplane Design from Viewpoint of Feeder Pickup Service," by A. B. Schultz. Presented before New York Section, Institute of Aeronautical Sciences, New York City, Sept. 15, 1944.)

"Small airplanes, such as the twin-engine Beech, with the lighter unit, will handle the smaller and newer routes where baggage pickup, and passenger requirements are moderate. Larger airplanes in the class of the Lockheed Lodestar or Douglas DC-2 equipped with the larger unit will adequately take care of the heavier and more established runs where mail and express pickups of 100 to 200 lb are expected and accommodations for a dozen passengers are needed.

"The present requirements for All American's proposed route extensions lie in the latter category around which the following general specifications are built. Similar specifications will apply to the smaller airplane.

1. The gross weight of the airplane should be about 15,000 lb to provide for 12 passengers and baggage, pilot and copilot, 300 lb of through express, 200 lb of pickup equipment, 1500 lb of pickup cargo capacity, and 500-mile range with the customary reserves.

2. The desired cruising speed is 175 mph with good control and pull-out characteristics at speeds as low as 120 mph for the pickups. It would be desirable to be able to cruise with good controllability at speeds as low as 80 mph in order to maintain operations when the visibility is reduced.

3. The powerplants should be adequate to permit the ship to operate out of 2500-ft sod fields and fulfill the requirements of CAR 04.75. In order to secure economical engine sizes with safety in case of one engine failure it may be necessary to obtain engines capable of 100% boost for emergency take-off, either by water injection or by "suicide" boost, as has been proposed by one manufacturer, as protection against this contingency. Addition of rocket

clusters in the tail of each nacelle for emergency boost requirements, operable in case of critical engine failure, may be a not too-distant solution to this problem.

4. To assist short take-offs and to slow down landings, high lift devices and reduced span loadings must be resorted to. To facilitate glide control for spot landings to minimize on-the-ground time, the use of spoiler-brakes would be most helpful. Shortened ailerons and built-in tip slots to maintain lateral control, coupled with greater aspect ratios to increase the relative flap length, offer interesting possibilities for improved performance.

5. The landing gear should be retractable and the ship in general kept as aerodynamically clean as possible to permit maximum economy at cruising speed.

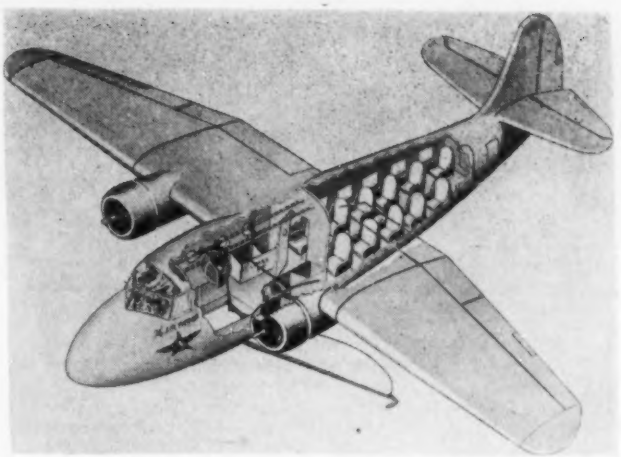
6. High wing construction, combined with tricycle landing gear, will provide excellent vision for pilot and passengers and provide maximum maneuverability on the ground. The tricycle gear will also permit moderate downwind landings and landings on single strip airparks in cross winds up to 20 mph.

7. Passenger comfort should equal that of intercity buses. The seats may be a few inches closer, nonreclining, and not quite as sumptuous as may be required on the mainline airplanes. A toilet should be provided. Adequate provision for airsickness should be readily available. Flying through low altitude turbulence and thermals, combined with the moderate pull-outs close to the ground, may result in more-than-usual passenger discomfort.

8. The passenger entrance should be aft of or under the wing to reduce the need for stopping engines at intermediate stops. Full flattening of propeller pitch, coupled with cowl cooling fans, would reduce discomfort to boarding passengers. The under wing location would protect the passengers in inclement weather when going to and from the limousine. It would seem most rational for the limousine, which has relatively plenty of time, to drive out to the airplane on the runway rather than have the airplane spend valuable time just taxiing to and from the passenger terminal, if any.

9. The pickup compartment should be located in the forward end of the ship so the copilot can act as pickup operator. Furthermore, it is desirable that the pickup operator be stationed as close to the pilot as possible so they may easily coordinate their functions. During take-offs, landings, and on instrument flights, the copilot can act as such. He can help the pilot when the pilot needs help, and whenever not needed "up front" act as pickup operator. Economical operation, perhaps the very existence of extensive air pickup and passenger service, will depend on adherence to this principle. It would be interesting to know the fare that would be required to support a crew of three – pilot, copilot, and pickup man, on a small, twin-engined plane capable of carrying only five passengers when full up! It is probable that the service would be so costly that very few passengers could afford its convenience.

10. The pickup unit itself should be located near the main spar to provide adequate tie-in structure for the pickup forces. This region is usually spoiled for passenger space anyway by the presence of the main spar and other necessary details, and provides an ideal spot for the pickup operator. The main mail bins will be located in the plane of the propeller discs and auxiliary bins installed wherever available space permits. About 150 cu ft of mail bin capacity is required.



■ Fig. 24 – Artist's conception of pickup-passenger airplane

11. The unit is mounted as high as possible over a hole in the floor so that cargo may be hauled all the way in by electric motor, the hatch closed, and the pouches sorted and recorded at leisure. The floor of the pickup compartment should be adjacent to the belly so that the raised pickup arm will be readily accessible to the operator's reach when resetting the hook for the next pickup.

12. A retractable guide pulley is located below the belly at the aft edge of the hole so as to guide the rope out during the pickup and to guide the container in during the wind-in, and also, to prevent a light container from banging against the belly of the airplane. The pulley retracts into the airplane with the container, guiding the latter in.

13. The contact arm is hinged forward of the hole, preferably below the pilot's cockpit so as not to reduce the capacity of the nose baggage compartment. This location for the arm reduces the distance the hook must be slid down the track and keeps the arm clear when heavy containers are hauled in. Mechanical actuation is recommended.

14. A locked door must be provided between the mail pickup and the passenger compartments.

"Fig. 24 was conceived by one of our artists and illustrates the high-wing principle. In this illustration the pickup unit is shown mounted on overhead rails to facilitate the handling of the cargo. This illustration was drawn several years ago and it is interesting to note how well many of our new, postwar airplanes will fit into this picture."

ERRATUM

The word "not" was omitted between the words "does" and "enter" in line 1, column 2, p. 679 of the article "Various Types of Compressors for Supercharging," by R. J. S. Pigott. The sentence should read thus: "It should be pointed out that real slip or leakage in rotary compressors is almost exclusively a function of pressure difference, since viscosity of air does not enter because the capillary seals are all 'rough,' and speed does not appear directly."

STRUCTURAL MO

THE AUTHOR: O. W. LOUDENSLAGER, assistant manager of the research and development department of Goodyear Aircraft Corp., joined that organization in 1928, the year following his graduation from Ohio State University. While design of rigid airships has been his forte, Mr. Loudenslager has also supervised the building and testing of non-rigid airships for the purpose of improving U. S. Naval patrol airships.

THE art of model building is said to date back to the ancient Egyptians, who buried model ships, called spirit ships, with their dead in order that the departed spirits could safely cross the sacred river Nile.

In modern times man has exhibited great ingenuity in designing, constructing, and testing models for the purpose of studying the stress and stability characteristics of indeterminate structures. For example, many models have been built in which the photoelastic properties of certain materials were used to determine stresses. These properties were discovered by Sir David Brewster[1] in 1816 and have been developed to a high degree by many other investigators.[2, 3] Rubber models in which studies have been made of the distortion under load of a grid or lattice of fine lines ruled on the surface have been useful for many purposes.[4] Brittle models made of such materials as plaster of paris or bakelite and tested to destruction have been used to predict the strength of structures and structural details.[5] Models in which deflections rather than loads are applied to determine influence lines mechanically from which stresses can be determined by the Müller-Breslau principle[6, 7] will be discussed more fully in a subsequent part of the text.

[This paper was presented at a meeting of the Buffalo Section of the SAE, Buffalo, N. Y., May 1, 1945.]

[1] See article on "Elasticity" in Encyclopaedia Britannica by H. Bateman, Vol. 8, p. 124, Fourteenth Edition, 1929, New York.

[2] See "Photoelasticity," by M. M. Frocht. Vol. 1. Wiley, New York, 1941.

[3] See NACA Technical Memorandum No. 824 (1937), "Photoelastic Investigation of Three-Dimensional Stress and Strain Conditions," by G. Oppel.

[4] See "An Experimental Study of Stresses in Masonry Dams," by K. Pearson and A. F. C. Pollard. Drapers Co. Research Memoirs. Technical Series V, Cambridge Univ. Press, 1907.

[5] See Univ. of Wis. Engineering Experiment Station Series Bulletin No. 81 (1935), "Predicting Strength of Structures from Tests of Plaster Models," by R. J. Roark and R. S. Hartenberg.

[6] See *Journal of Franklin Institute*, Vol. 203, 1927, pp. 375-386: "Use of Models in Solution of Indeterminate Structures," by G. E. Beggs.

[7] See "Strength of Materials," by S. Timoshenko. Van Nostrand, New York, 1940-1941.

[8] See *Journal of Aeronautical Sciences*, Vol. 6, October, 1939, pp. 499-501: "Notes on Recent Structural Research at Goodyear-Zeppelin Corp.," by K. Arnstein and E. L. Shaw.

Engineers at the Goodyear Aircraft Corp. have developed unique, and it is believed unusually precise, stress model designs to check theories on the stress analysis of the indeterminate rigid airship structure. The theoretical and model test results were found to be in excellent agreement. The purpose of this paper is to describe the design, construction, and testing procedure used in connection with these models.

Before describing the models developed at Goodyear Aircraft, the laws to which a model member must conform, if it is to represent the prototype accurately, will be presented.

■ Laws of Similarity

Although many stress models have been built and tested in which no particular attention was given to the relationship between the scales of dimensions

$$\left(N = \frac{\text{prototype dimensions}}{\text{model dimensions}}\right),$$

forces N_f, unit linear deformations N_d, and angular displacements N_θ, a proper relationship must exist if the most precise results are to be obtained without the use of correction factors which, in many cases, may be very difficult to evaluate with certainty.

The outstanding feature of the models about to be described is that the relationship between these scales can be varied almost at will and the optimum values easily obtained. Before discussing the manner in which the proper relationship has been accomplished, the relationship itself will be considered.

The fundamental rule of similarity as applied to stress models is considered axiomatic and may be stated as follows:

In order that a model may faithfully represent the prototype in both first- and second-order effects, it must be

AL MODEL TESTING

by O. W. LOUDENSLAGER
Assistant Manager
Research and Development
Goodyear Aircraft Corp.

IT is the purpose of this paper to discuss in detail unusually precise structural model design, construction, and test procedure.

A statement of the laws of similarity to which all structural models must conform if precise results are to be obtained in the simplest manner is given by Mr. Loudenslager. Scale selection, choice of materials, and construction methods are all considered by him.

Three model members of designs which can represent a number of prototype properties are described in detail, as well as the use of each of these members, and the determination of bending, torsional, tensile, and compressive stresses by means of simple equipment. In addition, design formulas are set forth for a member which represents the axial, torsional, and bending (in two planes) properties.

Finally, three test procedures, all applicable to a variety of simple or complicated indeterminate structures, and the methods used in evaluating the test results are presented by the author.

Briefly, these are: tests in which influence lines are photographed so that axial and bending stresses in redundant structures can be determined; stability tests for determining buckling strength; and tests to give bending and other stresses.

geometrically similar to the prototype both before and after loading.

Hence, from geometry:

$$N_d = 1 \tag{1}$$

and:

$$N_\theta = 1 \tag{2}$$

N_l can be varied within limits but:

$$N_l = N^2 \tag{3}$$

has been favored for the Goodyear Aircraft models and will be used throughout this paper. This is the relationship that would exist if the model member were a scaled down replica of the prototype. It has been found satisfactory in a wide variety of tests. N may have any convenient value and will be discussed more fully in the section entitled "Choice of Scale." If it is assumed that the model joint stiffness is approximately to scale, the axial, bending, and torsional stiffness of the model girder must then be as follows:

$$EA = D_a = \frac{D_{ap}}{N^2} \tag{4}$$

$$EI = D_b = \frac{D_{bp}}{N^4} \tag{5}$$

$$GJ = D_t = \frac{D_{tp}}{N^4} \tag{6}$$

in which D is stiffness and subscripts a, b, and t refer to axial, bending, and torsion, and p refers to the prototype. Model forces, both external and internal, such as the initial tension of wires and cables, are represented by:

$$F = \frac{F_p}{N^2} \tag{7}$$

In models subjected to fluid pressure both the pressure and the area vary as N^2, hence model pressure:

$$P = P_p \tag{8}$$

From the rule of similarity previously given, it is apparent that the principle of superposition is strictly correct only as long as the load displacement relationship is linear. Examples of nonlinearity which tend to produce errors with the use of superposition are first, deflection effects which cause changes in the action of the external or internal forces, and second, the use of materials with nonlinear stress-strain relationships. The principle of superposition was checked on a model of a complete airship hull structure[8] by applying first, axial forces, then side forces, then the combined forces. It was found that for

■ Fig. 1 – Model of nonrigid airship for water model tests

such a highly redundant structure no serious error would result in the use of this principle.

It is not always possible to conform to the above rule of similarity, and if a deviation is made, it should be considered when evaluating the results. One good example of this is found in the water model tests of nonrigid airships[9, 10, 11] which have been conducted at Goodyear Aircraft. See Fig. 1. In these tests, models of the airships were built complete with internal and external suspensions, control car, and fins. The models were filled with water and suspended in an inverted position from the center of gravity of the control car. Proper gravitational and simulated aerodynamic loads were applied and various characteristics of the suspensions and envelope studied. Since the stress-strain relationship of fabric is nonlinear and since the properties of fabrics vary widely one from the other, it was necessary to build the envelopes of the same fabric which was to be used in the prototypes and to arrange the force scale N_f so that prototype stresses were applied to the model envelopes. In order to secure proper pressure gradient, the dimensional scale N was set equal to 31.6.

■ Design of Structural Model Members

It is usually impractical to make a structural model member a scaled-down replica of the prototype as would be required by a strict application of the rule of similarity. Fortunately, however, this is not necessary because if the elastic properties of the model members conform to equations 4, 5, and 6, it is obvious that the centroids of the members will abide by the rule. Hence, *model girders may have any practical shape which gives them correctly scaled elastic properties.* When external forces are to be applied to the structure between joints and the members are not scaled-down facsimiles of the prototype, care must be exercised in designing the model member so that the applied forces cannot produce out-of-scale deflections.

A number of models whose members accurately represent the elastic properties of the prototype have been built and tested at the Goodyear Aircraft Corp.[8, 12-20] One of the simplest model girders of this type is shown in Fig. 2. This girder can be designed to represent correctly the axial and bending properties of the prototype in one plane. The EI of the straight part L' was designed to give the model the correct bending stiffness, and the two parts S were designed to give the proper axial stiffness. By properly choosing the width and thickness of the L' part of this member and at the same time keeping the value of I constant, it can usually be made correctly to represent the torsional elasticity of the prototype. Although members which correctly reproduce a greater number of elastic properties are now available, this member is still useful for certain purposes, as will be evident subsequently.

Model girders of the type shown in Fig. 3 were designed to represent accurately the axial stiffness and bending stiffness in the plane of the paper and to represent approximately the torsional stiffness of the prototype. These members indicated axial load by the rotation of mirrors A together with bending and torsion by the rotation of mirrors B. Each of them consisted of two Z-shaped members which were connected by the joint block to which the end mirrors B were attached. The longitudinal members were split tubes connected at the center by cross pieces to which mirrors A were attached. These cross pieces were quite rigid, so that the girder reacted to bending and torsion in practically the same manner as two split tubes. The axial stiffness was controlled by regulating the length of the cross pieces. Members of this design can be used only when loads are to be applied to the joints, because loads applied directly to the girder would affect the relative rotation of mirrors A. The design theory for girders of this type is given by Donnell, Shaw, and Potthoff.[15]

A model girder which still more accurately represents the elastic properties of the prototype is shown in Figs. 4, 5, and 6. Fig. 7 shows a complete model made of girders of this type. The member consisted of two sinuated side plates which were milled out of yellow brass sheet having a yield point of 38,000 psi and a modulus of elasticity of 14.7×10^6. The side plates were connected by spacers having an H-shaped cross-section which was milled out of yellow brass bar. The girder was soldered together as shown, and it was equipped with mirrors by means of which axial, bending, and torsional stresses could be measured. Bending and torsional stresses were determined by

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[9] See Goodyear Aircraft Corp. Report No. R-55-41, Part II, "Hydrostatic Investigation of Nonrigid Airship," by O. W. Loudenslager and R. C. Gardner. Unpublished.

[10] G. A. Crocco in periodical "La Technique Aéronautique," Paris, June 1, 1911.

[11] See "Pressure Airships," by T. L. Blakemore and W. W. Pagon. Ronald Press, New York, 1927.

[12] See "Application of Stress Models to Specific Structural Problems," by S. F. Tingley. Experimental Stress Analysis, Vol. 3, No. 1.

[13] See *Aeronautical Engineering Review*, Vol. 1, June, 1942, pp. 13, 15, 17: "Recent Developments in Airships," by K. Arnstein.

[14] See "Methods of Calculating Stresses in Hulls of Rigid Airships," by K. Arnstein and E. L. Shaw. Proceedings of Fifth International Congress of Applied Mechanics, Cambridge, Mass., September, 1938.

[15] See ASME Transactions (Applied Mechanics), Vol. 60, A-67-A-77: "Stress Model of Complete Airship Structure," by L. H. Donnell, E. L. Shaw, and W. C. Potthoff.

[16] See "Model Measurements and Airship Stress Analysis," by L. H. Donnell. Publication No. 3, Daniel Guggenheim Airship Institute, Akron, Ohio, July, 1935.

[17] See "Some Phases of Structural Research at Goodyear Aircraft Corp," by E. L. Shaw. Experimental Stress Analysis, Vol. 1, No. 2, 1944, pp. 90-100.

[18] See Goodyear Aircraft Corp. Report No. R-55-34, Parts I, II, III, "Elastic Stability Model Test of Airship Hull Structures between Main Frames," by E. L. Shaw and O. W. Loudenslager. Unpublished. U. S. Navy Contract No. 47423.

[19] See U. S. Patent No. 2,205,102 issued to O. W. Loudenslager and L. H. Donnell on June 18, 1940, for "Model Test Piece."

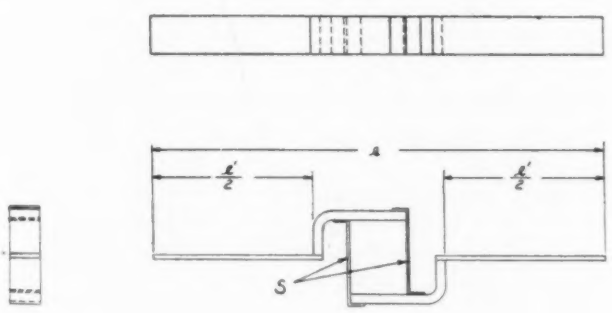
[20] See Goodyear Aircraft Corp. Report No. R-55-44, Part I, "Theory for Design of Structural Models," by O. W. Loudenslager. Unpublished.

measuring the angular rotation of mirrors attached to the joint blocks (Fig. 6) employing the usual method of reading the mirrored image of a scale with a telescope. The center mirrors were mounted on a hinge (Figs. 4 and 6). Rods connected the halves of the hinges to the ends of the girder in such a way that changes in length caused the mirrors to rotate. The change in angle was determined as before and converted into axial stress. A spring was provided to eliminate lost motion. The design of this member was such that loads could be applied directly to the member without affecting the stress indications given by the rotation of the mirrors.

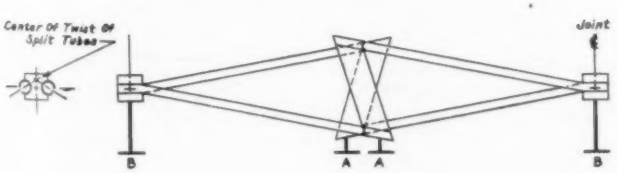
This girder can be designed to represent accurately the axial stiffness, bending stiffness in two planes, and torsional elastic properties of the prototype. The relationship between the axial stiffness and the bending stiffness on the X-X axis is controlled by the radius R and the desired values obtained by providing the proper moment of inertia $I = t\,h^3/6$. By varying a, b, c, t, h, and s the desired Y-Y bending and torsional stiffness can usually be obtained.

It is not always possible to get the desired values for both of these properties, but one of the two can be made exact and the other approximate. In the Goodyear Aircraft models there was a tendency for the torsional stiffness to become too large when the Y-Y bending was made correct. For the models tested this was not serious because torsional stiffness did not play an important part. It is believed that ordinary structural members would give less trouble than airship girders in this respect because their torsional stiffness is usually larger with respect to their bending stiffness. For example, in a rolled or extruded section of the usual type, the polar moment of inertia is $J = I_X + I_Y$, but the effective J of an airship girder is somewhat less than this due to the trussed construction.

The theory for the design of this model member has been developed in detail by the author[20] and the results carefully checked by the calibration of model members. The resulting design formulas are as follows:



■ Fig. 2 – Simple type of model girder



■ Fig. 3 – Model girder designed accurately to represent axial stiffness and bending stiffness in plane of paper and approximately to represent torsional stiffness of prototype

■ Notation

The following notation is used throughout this paper. All symbols refer to the model unless marked with subscript p which indicates prototype. For symbols not shown refer to Fig. 5.

- A = Area
- D = Stiffness
- E = Modulus of elasticity
- F = Force
- G = Shear modulus of elasticity $= \dfrac{E}{2.6}$
- I = Moment of inertia
- J = Polar moment of inertia
- k = Constant
- l = Length
- M = Moment
- N = Scale $= \dfrac{\text{Prototype dimensions}}{\text{Model dimensions}}$
- n = Number of semicircles in length of girder between joints divided by two
- P = Unit of pressure – also load
- p = Subscript denoting prototype
- t = Subscript denoting torsion
- x, y = Subscripts denoting axis of bending, see Fig. 5
- β = See Table 1 (Taken from Timoshenko[7])
- δ = Axial elongation
- θ = Angular rotation

$$A_p E_p = \frac{t\,h\,E\,N^2}{\dfrac{3\,\pi\,R^2}{2\,h^2} + 1.14} \tag{9}$$

$$E_p I_{pX} = \frac{t\,h^3\,E\,N^4}{3\,\pi} \tag{10}$$

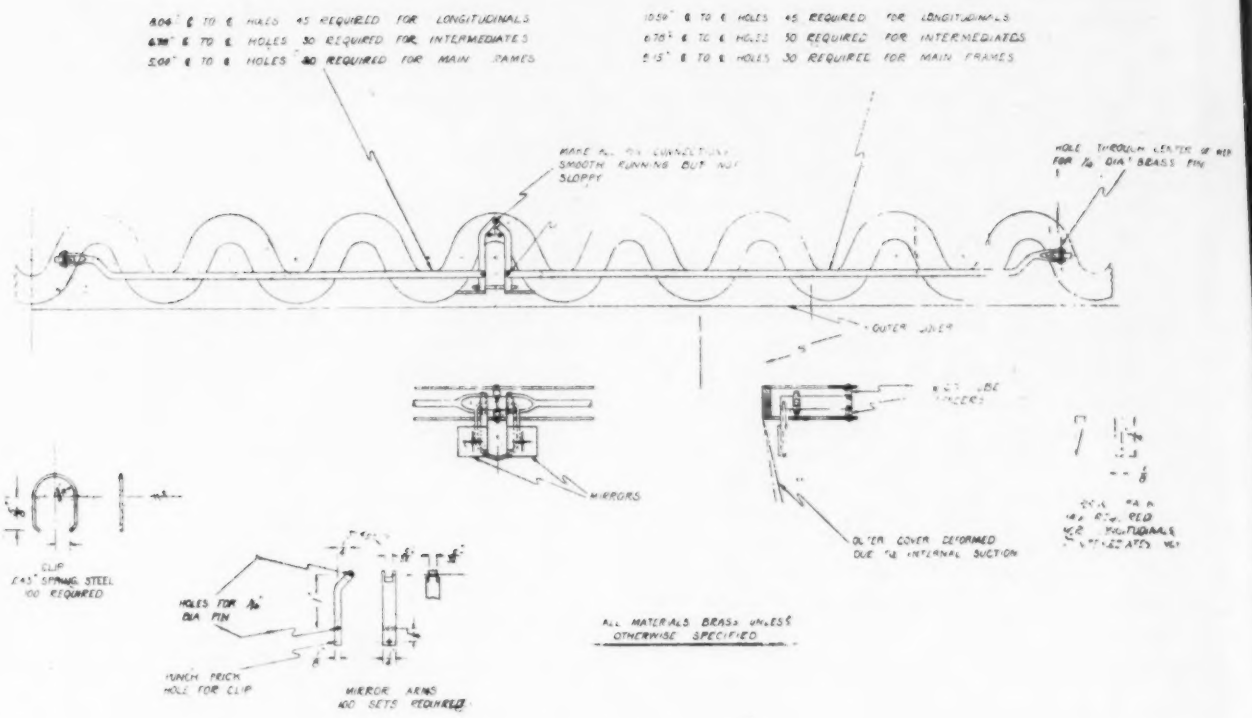
$$I = \frac{t\,h^3}{6} = \frac{\pi\,E_p\,I_{pX}}{2\,E\,N^4} \tag{11}$$

$$R = \sqrt{\frac{2\,I_{pX}}{k_1\,A_p\,N^2}} \tag{12}$$

where:

$$k_1 = 1 + \frac{h^2}{4.14\,R^2} \tag{13}$$

The value of $k_1 = 1.18$ may be used as an average value for a first approximation. Girders used at the Goodyear Aircraft Corp. varied between $k_1 = 1.11$ and $k_1 = 1.24$.


MIRRORS
OUTER COVER DEFORMED DUE TO INTERNAL SUCTION
ALL MATERIALS BRASS UNLESS OTHERWISE SPECIFIED
MIRROR ARMS 400 SETS REQUIRED

■ Fig. 4 – Axial strain instrument

$$E_p I_{pY} = \frac{s^2 E N^4}{k_1 R}\left(\frac{\beta b c^3}{2.6 a} + \frac{t h^3}{6 \pi R} + \frac{k_1 k_2 t h^3}{R}\right) \quad (14)$$

where k_2 is a constant depending upon the end conditions. When the girder is pin connected, $k_2 = 0$ and the term drops out. For the Goodyear models $k_2 = 0.013$.

Table 1

$\frac{b}{c}$ =	1.00	1.50	2.00	3.00	4.00	6.00	8.00	10	∞
β =	0.141	0.196	0.229	0.263	0.281	0.299	0.307	0.313	0.333

$$G_p J_p = E N^4\left[\frac{s^2}{\frac{5.2 a R}{\beta b c^3} + \frac{48 \pi R^2}{t h^3}} + \frac{h t^3}{7.92} + k_3\left(\frac{s^2 t h^3}{16 l^2} + \frac{\beta s^2 b c^3}{a l}\right)\right] \quad (15)$$

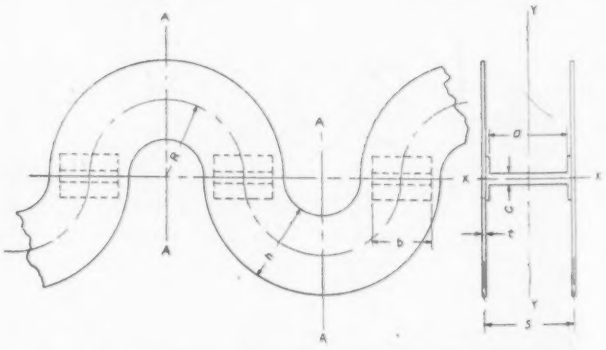
where k_3 is a constant depending upon the end conditions. When the girder is pin connected, $k_3 = 0$ and the term drops out. For Goodyear models $k_3 = 3.2$.

Area of a model wire is:

$$A = \frac{A_p E_p}{E N^2} \quad (16)$$

The joint must be located at section *A-A*, Fig. 5, to avoid undesirable rotation when axial loads are applied. If $\frac{l}{4R}$ is not an integer it is necessary to "stretch" the girder slightly. The axial stiffness and Y-Y bending stiffness should then be multiplied by:

$$\frac{}{4 R n} \quad (17)$$



■ Fig. 5 – Details of girder

■ Fig. 6 – Photograph of model girder

where $2n$ is the number of semicircles in the length of the girder.

■ Choice of Scale

The length scale should be chosen for convenience and economy because very accurate results are possible from either large or small models. If a model were too large, ladders and scaffolding would be required to take readings and the application of dead weight loads might involve considerable labor. If extremely small, it would be very tedious and, therefore, expensive to build. The scales for the models built at Goodyear Aircraft have usually been taken between $N = 10$ to $N = 30$. The scale of the largest of these models was chosen so that certain special instruments could be used in connection with the tests, and also to permit a man to make observations from within the model.

■ Construction

Many different materials have been used by numerous investigators for the construction of stress models, but metals are preferred for the models under consideration. Although plastics are excellent for some kinds of models because of their photoelastic properties, they are often seriously affected by aging, temperature, humidity, and a tendency to creep under load. Metals are excellent building materials for stress models because of their resistance to aging and humidity. Temperature effects can be neutralized by using the same metal for the entire model. Properly chosen metals will not be appreciably affected by creep under load.

The model building material most used at Goodyear Aircraft is hard drawn or spring tempered yellow brass and similar high strength nonferrous metals. In addition to the good properties which these materials share with other metals, they have good corrosion resistance, low modulus of elasticity, easy machinability, and good solderability.

Although model members have been successfully made of cold-worked materials which cannot be stress relieved without drastic reduction in strength, it is considered good practice to avoid adding the "locked up" stresses caused by bending, stamping, or shearing the parts. They should rather be cut on a milling machine or with other similar equipment.

The manner in which a model should be assembled varies with the type. Assembly methods should receive careful consideration both before and during the design in order to achieve the proper degree of accuracy. Care should be exercised to avoid excessive eccentricity, remembering that model eccentricity should be $1/N$ of the prototype value in order to conform to the rule of similarity. In general, the models described in this paper were assembled in carefully made wooden jigs. Wood was chosen for its inexpensiveness and because its low heat conductivity made soldering the joints easier. In soldering it is advisable to avoid overheating the metal and drawing the temper.

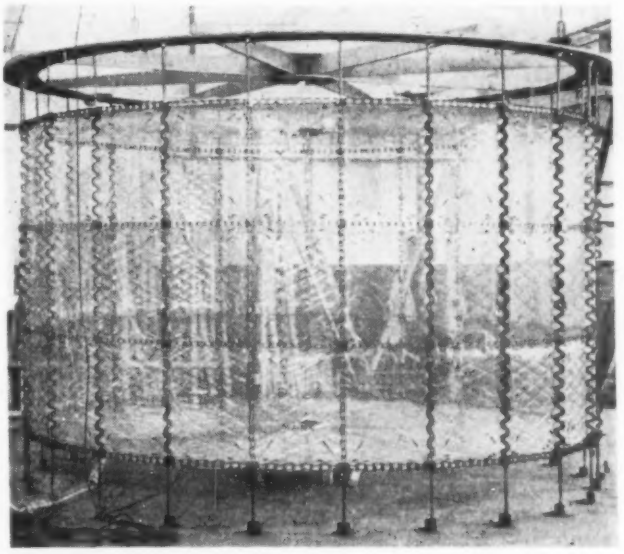
Model members should be calibrated before assembly to ensure proper stiffness. When preparing the Goodyear model girders for calibration, joint blocks, of the exact designs that were to be used in the model, were attached to the ends, the distance between them being the same as the length of the model girder. In this way the effect of the end conditions (see equations 14 and 15) was automatically taken into consideration.

The axial stiffness calibration was accomplished by applying axial forces in reasonable increments, reading the rotation of the center mirrors, and registering the change in length with a dial gage.

The calibration of the X-X and Y-Y bending and torsional stiffness was made by applying couples to the ends and noting the rotation of the end mirrors. Pure bending and torsion was applied in this manner to avoid possible errors due to shear deflection.

■ Testing

The testing methods used may be divided into two classifications; the indirect in which influence lines were first determined from which stresses were later obtained, and the direct in which scaled-down loads were applied directly to the models.



■ Fig. 7 – Complete model

The design of a model for indirect testing is fundamentally the same as for direct testing in that both should correctly represent the stiffness of the prototype. The model members must be capable of being bent and unbent if bending moments are to be determined. Members of the type shown in Fig. 2 have been used at Goodyear for this purpose. If large deflections are expected, tension members, such as cables and tie rods, may be represented by coil springs.

The indirect testing method described below is based on Müller-Breslau's application of C. J. Maxwell's reciprocal theorem,[21] which states: *In an elastic body, the work done by the forces of the first state on the corresponding displacements of the second state is equal to the work done by the forces of the second state on the corresponding displacements of the first.* For proof of the theorem see Timoshenko.[7] The late Prof. G. E. Beggs[6] was first to draw attention to this type of testing.

Figs. 8 and 9 illustrate an indirect test on a structural model. The model was supported on pin points and rested against a piece of very slow photographically sensitive paper, such as ozalid paper, which could be developed relatively dry to avoid shrinking. Before making the exposure a clamp was applied at a point where it was desired to determine the influence lines for bending. This clamp bent the model through a known angle $\theta/2$. After an exposure of approximately one-half full time the clamp was removed and a reverse bend put on the model member at the same point to balance out large deflection effects as much as possible. The exposure was completed and the paper developed, giving the influence lines shown in Fig. 9. In order to determine the influence lines for an axial load, the member was cut and separated a known distance d_o, avoiding rotation or lateral displacement. The above procedure was repeated for each point where axial and/or bending stresses were to be determined.

In order to evaluate the axial load, the displacements $(d_1, d_2, \ldots\ldots d_n)$ of the influence lines of the model in the direction of the corresponding prototype applied loads $(P_1, P_2, \ldots\ldots P_n)$ were measured with a low power microscope which had a scale in the eyepiece. Then from Maxwell's theorem:

$$P_p\, d_0 = \Sigma\,(P_{p1}\, d_1 + P_{p2}\, d_2 + \ldots\ldots P_{pn}\, d_n)$$

and the prototype axial load P_p at the point of separation was:

$$P_p = \frac{\Sigma\,(P_{p1}\, d_1 + P_{p2}\, d_2 + \ldots\ldots P_{pn}\, d_n)}{d_0} \qquad (18)$$

Similarly, the prototype bending moment at the point where the clamp was applied equals:

$$M_p = \frac{\Sigma\,(P_{p1}\, d_1 + P_{p2}\, d_2 + \ldots\ldots P_{pn}\, d_n)\, N}{\theta} \qquad (19)$$

where N is the scale factor as previously defined. It should be noted that no scale factor is needed in equation 18.

In model testing of this type, the model does not conform to the rule of similarity previously stated, and for this reason second-order effects are not accurately taken into account. Another possible source of error is that due to large deflection effects. The importance of these can be determined experimentally by testing the model with both large and small displacements d_o and θ. If the large deflection effects are unimportant the results will be practically identical.

The method described above is particularly valuable for investigations of planar structures in which second-order effects are not important. It has also been very successfully used on continuous-beam problems, but it is not well adapted to three-dimensional structures.

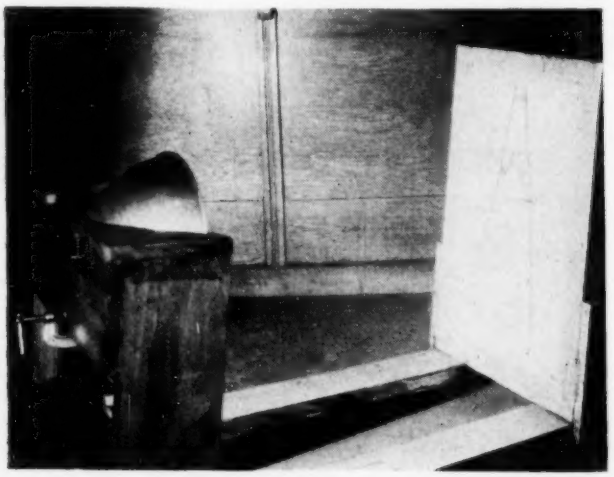
In direct testing, scaled-down loads are applied to the model. Such tests may be divided into two classes; namely, stability tests and stress tests.

In making stability tests it should be pointed out that since the buckling strength of a structure depends not upon its strength but rather upon its elastic properties, it is very important that the elastic properties be faithfully represented in the model. For this type of testing it has been the practice at Goodyear Aircraft to load to destruction models which incorporate all of the major structural features of the prototype.

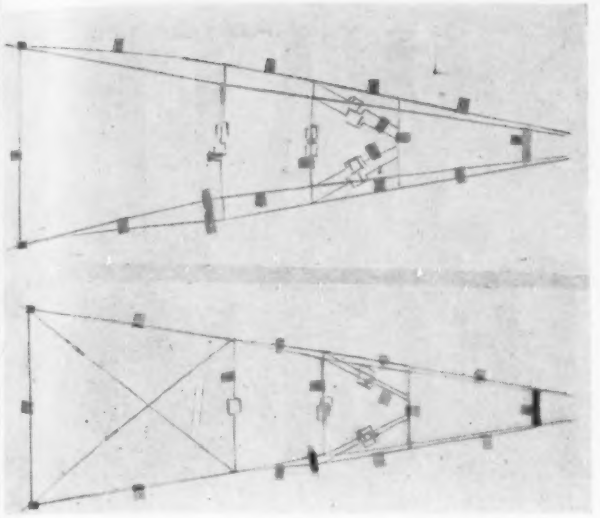
Since it is desirable to have the model fail in stability, rather than by yielding, it is necessary to make the best possible estimate of the greatest load to which the members will be subjected at failure. They are then designed to a strength well in excess of this value. This may be accomplished by using a fairly high strength material, and since it is difficult to predict the strength accurately, single members should be tested before building the entire model.

The elastic stability model tests of an airship hull structure between main frames[18] conducted at the Goodyear Aircraft Corp. are examples of this type of testing. In these tests a model with scale $N = 30$ was built of a center bay of a rigid airship. The girders were similar to those illustrated in Figs. 4 and 5 except that they were smaller and were made in one piece by milling them out of a sheet of yellow brass (yield point 38,000 psi). The wires

[21] See "Elastic Arch Bridges," by C. B. McCullough and E. S. Thayer. Wiley, New York, 1931.



■ Fig. 8 – Indirect test on structural model



■ Fig. 9 – Influence lines

were made of spring temper brass with a tensile strength of approximately 150,000 psi.

One-half of this model was loaded so as to correspond to the compression loads resulting from bending. Radial displacements of the joints were measured and a not entirely successful attempt was made to use Southwell's method[7] to predict the failing load. The radial displacements were also used to determine the wave pattern of the structure prior to failure. After failure, the model was repaired and the tests repeated for another condition. The results from these tests compared very favorably with those obtained from stability theory.

When applying direct testing methods to determine stresses in the members, it is not necessary to load the model to failure because adequate means are available to measure accurately the deflections and hence the prototype stresses. It is not even wise to load the model to failure because the yield strength of various parts is not necessarily to scale. Tests made on the model shown in Fig. 7 are an example of this type of testing.

After loads were applied, axial forces were measured by observing the relative rotation of the two center mirrors and determining the change in length of the girder from the known dimensions of the hinge, Fig. 4, and the calibration data. The prototype axial force is:

$$P_p = \frac{\delta A_p E_p}{l} \qquad (20)$$

The prototype torsion moment was measured by reading the relative rotation of the mirrors at the end of the girders and applying the relationship:

$$M_{tp} = \frac{\theta G_p J_p}{l N} \qquad (21)$$

Average prototype bending moments were measured by observing the relative rotation of the mirrors at the ends of the model member and also the relative rotation between each end and the average of the center mirrors. The average bending moment was then:

$$M_p = \frac{\theta E_p I_p}{l N} \qquad (22)$$

where l in this case was the distance between the points of attachment of the mirrors to the girder. If more detailed bending information were required, additional mirrors could be attached to the girders at section *A-A*, Fig. 5. The mirrors used for this purpose *must not* be attached to the spacer blocks which rotate when axial, *Y-Y* bending, and torsion loads are applied.

Wire tensions were measured by a special tensiometer which was constructed to deviate the wire from a straight line and measure the component of its tension. The instrument was calibrated to read tensile forces directly for each of the wire sizes used in the model.

■ Discussion

The accuracy with which general prototype loads can be determined by use of the direct model testing procedure outlined in this paper is probably greater than that which could be obtained when testing a full-size structure. The reason for this is that it is often difficult to distinguish between general and local loads in full-scale tests, whereas models of the type described indicate general loads only. The degree of perfection depends entirely upon the care used in designing, constructing, and testing the model. Careful investigation in determining the resultant of all forces measured at a joint and dividing the "closing force" thus determined by the yield strength of one of the larger members indicated that test results were accurate to within approximately 5%.

It is believed that these models can be tested dynamically by substituting properly placed electric strain gages for the mirrors shown in the illustrations. When making such tests it should be borne in mind that the mass of the model is out of scale. It is in the order of four times the equivalent weight of the prototype.

Although the dead weight of a model of the type described is not to scale, the stresses due to dead weight may, of course, be determined by applying simulated dead weight loads and reading the differences between the displacements at these loads and zero.

No serious attempt has been made to produce girders with scaled-down yield points. In order to avoid danger of failure the model girders were designed to have yield strengths considerably in excess of the prototype scaled down and the yield strength of the structure was determined as previously given.

The design of model girders in which many or all of the elastic properties of the prototype are correctly duplicated is in its infancy. It has far reaching possibilities in the art of structural model testing. The Goodyear models have been primarily used for investigations of rigid airships but it is believed that model testing of the kind presented can be applied to many other types of structures. There is little doubt that further improvement in models embodying the fundamental principles presented in this paper will be forthcoming.

THE AUTHORS: M. R. ROWE (J '41), assistant project engineer of Wright Aeronautical Corp., handles problems concerning water injection systems, engine power controls and related items. Mr. Rowe came to Wright in 1940, shortly after graduating from New York University. G. T. LADD (J '42) has been with the engineering department of Wright Aeronautical Corp. since his graduation from Case School of Applied Science in 1938. He is now assigned to the design and development section of the engineering department, responsible for projects dealing with water injection systems.

WATER INJECTION

THE use of water and water-alcohol injection as an engine internal coolant has a considerable background that has been very ably reviewed in papers already presented to the Society of Automotive Engineers and to other engineering organizations.

Reports of its use at the Wright Aeronautical Corp. date back to November, 1933, when the investigation of 80- and 87-octane fuels for take-off power output included the use of water injection to prevent overheating. The investigation concluded that take-off operation with 87-octane fuel and water could be run indefinitely, that the use of 80-octane fuel with water coolant was marginal, and that water additions up to 40% by weight of fuel did not adversely affect power output or speed. Again in 1935, during an attempt to induce carburetor throttle icing, the benefits of water injection were noted in the resultant smooth engine operation and reduced operating temperatures.

Ford L. Prescott has given data showing large increases in engine permissible power output when water was used to inhibit detonation.[1] Other investigators reported progress in the work of evaluating the merits of water and water-alcohol mixtures as antidetonant fluids, and Hives and Smith have discussed briefly the increase of knock-free bmep when water injection is used.[2]

With the advent of international conflict, and the pursuant rush to get something out quickly which would increase available horsepower, applications of water injection systems for aircraft engines were hurriedly designed and put into service. Background experience was theory only, and misuse common.

The major considerations of water injection system design were:

1. Excess power must not be available unless water was flowing into the engine.

2. The power must immediately be returned to a safe dry limit when the water supply was exhausted.

The quantities of water or water-alcohol metered to the engine were determined on the basis of experimental testing, and limited by detonation suppression and maximum permissible cylinder-head temperatures.

The "art" has progressed considerably since those early days, and today we have a fairly complete understanding of water injection application and limitations.

This paper, although titled "Water Injection for Aircraft Engines," deals with water-alcohol mixtures as well as straight water. The term "antidetonant injection" will be used synonymously with that of "water" or "water-alcohol" injection.

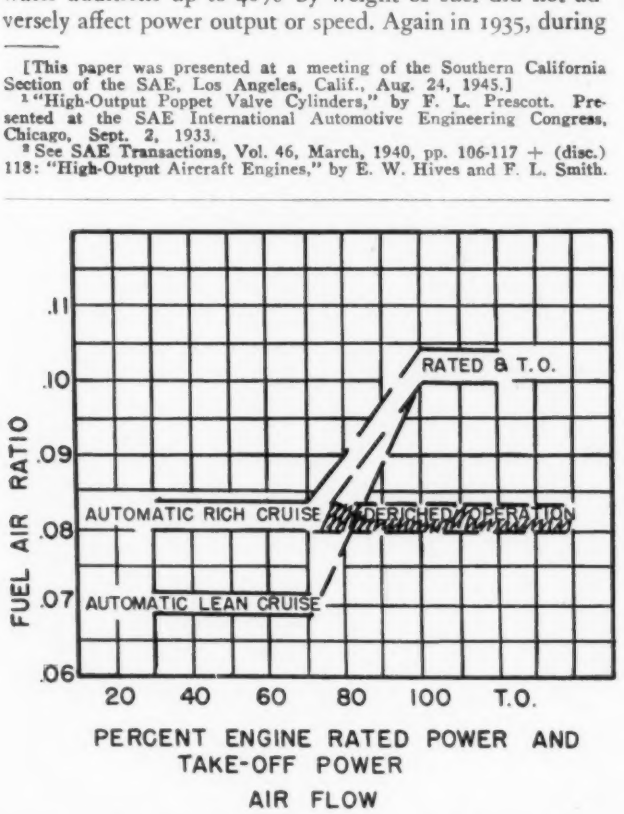
This paper will not attempt to establish the theories involved in the application of water injection, but rather to illustrate the results or effects of water and water-alcohol injection on aircraft-engine performance.

Basically, calculations have shown that the cylinder-head temperature cooling obtained using water or water-alcohol injection represents approximately 30 to 40% of the available heat of vaporization of the injected water. However, this heat of vaporization is not considered to be the total resultant effect of water injection. Test result calculations on the compression of gaseous fluids have demonstrated

[This paper was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Calif., Aug. 24, 1945.]

[1] "High-Output Poppet Valve Cylinders," by F. L. Prescott. Presented at the SAE International Automotive Engineering Congress, Chicago, Sept. 2, 1933.

[2] See SAE Transactions, Vol. 46, March, 1940, pp. 106-117 + (disc.) 118: "High-Output Aircraft Engines," by E. W. Hives and F. L. Smith.



■ Fig. 1 – Typical, aircooled radial engine fuel-air requirements

for Aircraft Engines

by M. R. ROWE and G. T. LADD
Wright Aeronautical Corp.
Division of Curtiss-Wright Corp.

THE main purpose of this paper is to illustrate the results or effects of water and water-alcohol injection on aircraft-engine performance.

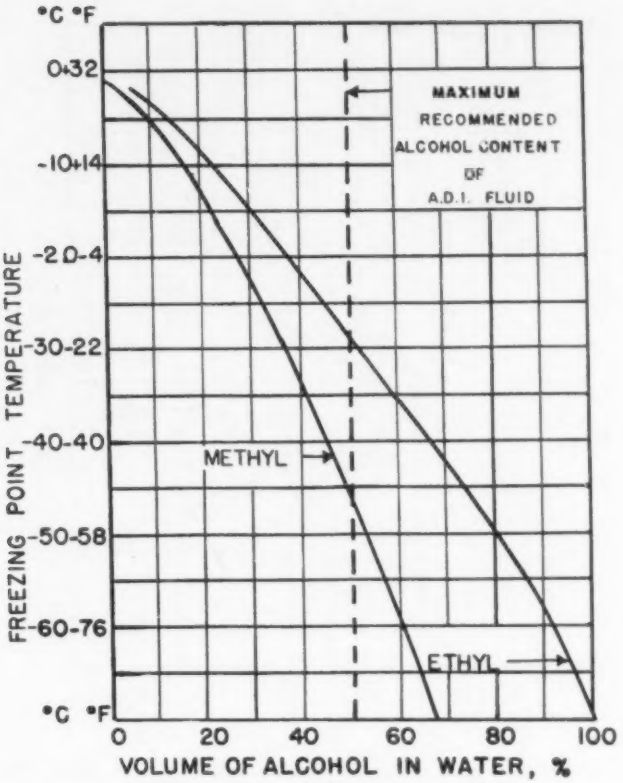
Water offers the best rate of cooling at high power output and, the authors point out, should be used without any additive where cooling is of paramount importance.

Water-methanol mixtures, on the other hand, they explain, will afford the most increase in power output normally limited by destructive detonation.

Because of the increased flexibility possible with the use of such fluids, military installations have already taken advantage of them, and commercial operators will want to study the problem thoroughly to determine how their use will fit into the postwar plan of operations.

that the work exponential of the basic equation $PV^n = K$ will be reduced as water is injected into the work cycle, and that this reduction may amount to 10-12%. This change indicates that the specific heat during the combustion cycle would have to be taken into account in any truly theoretical analysis of water injection. Too, calculations made at absolute intake charge pressure conditions, for constant speed and increasing horsepower at constant cylinder-head temperatures, show that the slope of specific brake water consumption does not change when going from a pressure condition of complete vaporization to that where the pressure is too high to permit vaporization. This is important and indicates that the major cylinder-head temperature cooling effect is obtained during the high temperature and pressure peaks of the combustion cycle, since it does not appear to matter whether the water enters the cycle as either vapor or liquid.

General experience at the Wright Aeronautical Corp. has taught that water and water-methanol mixtures are best suited for detonation inhibitors and internal coolants. The addition of 50% methanol to water by volume is considered to be optimum for water-methanol mixtures. Tests with pure alcohol, either methyl or ethyl, have indicated decreases in detonation limited power outputs as compared to 100-octane fuel. Investigation of water vapor injection into induction air ahead of supercharging has shown that this medium is an effective detonation inhibitor at lean fuel-air ratios only. Many other fluids have been evaluated as internal coolants and detonation inhibitors. This paper, however, will be devoted to the application of water and water-alcohol additions only.



■ Fig. 2 – Freezing points of methyl and ethyl alcohol solutions

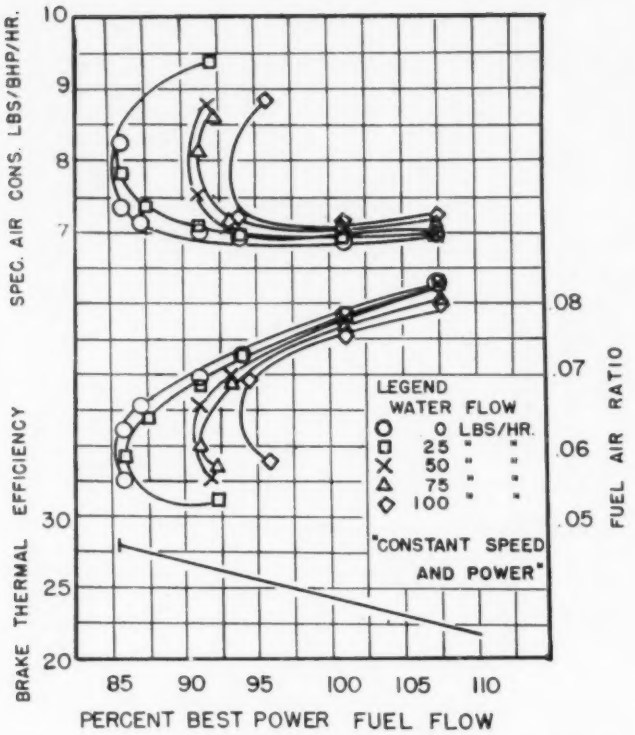
■ Data and Test Results

A. Introduction.

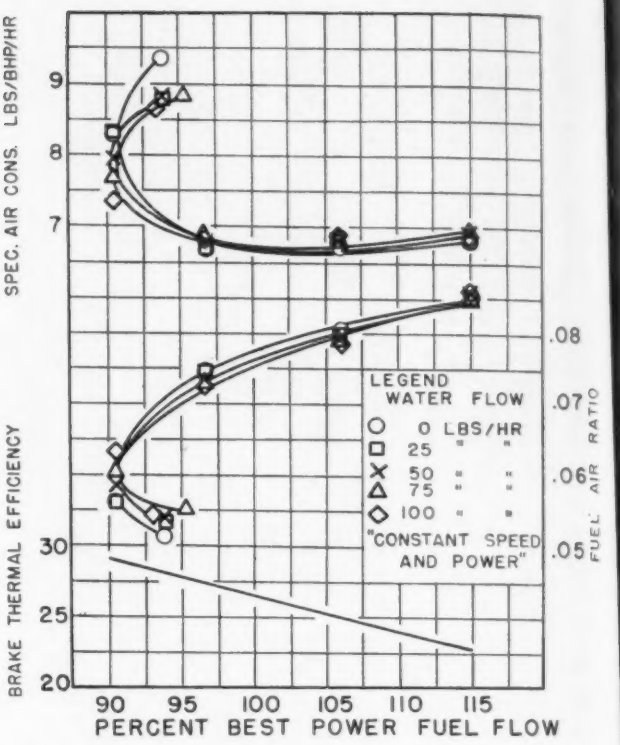
1. Fuel Requirements – To permit a better understanding of the data presented in this paper, Fig. 1 has been drawn to illustrate typical, radial aircraft-engine fuel-air requirements. Carburetion fuel-air limits are shown, since this paper deals primarily with water injection results obtained with carburetion engines.

The automatic rich and automatic lean limits both lead to a single rated and take-off fuel-air range. The deriched fuel-air limits shown are normally chosen for derichment operation because they are closest to engine best power (the most brake horsepower output for the least indicated horsepower output), and are available for use by merely restricting the jet flow used for rated, take-off, and powers in excess of take-off.

Normal cruise operation is performed in automatic lean from approximately 30 to 70% of rated power. Any in-



■ Fig. 3 – Effect of water injection on engine brake thermal efficiency at low cruise power – Wright Cyclone 9 engine



■ Fig. 4 – Effect of water injection on engine brake thermal efficiency at high cruise power – Wright Cyclone 9 engine

crease in power above 70% of rated is accompanied by enriched mixtures to provide for engine cooling at high cruise powers and to prevent detonation, which is generally caused by local overheating within the combustion chamber.

The use of fuels of lower octane value than those for which the engine was designed is usually compensated for by enriching the carburetor mixtures. Aircraft cooling installations which do not provide sufficient baffle pressure drop for proper engine cooling are sometimes compensated for by increasing operating fuel-air ratios.

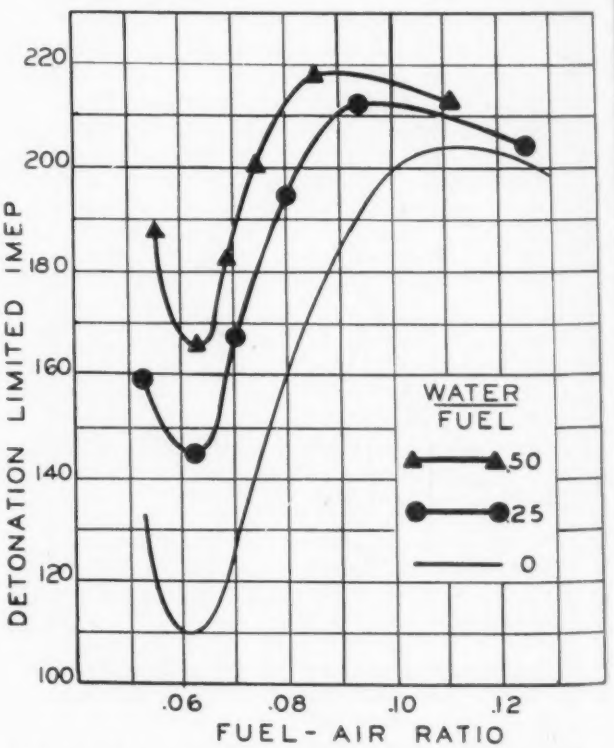
2. *Antidetonant Fluids* – Selection of the antidetonant fluid as a supplementary addition to basic fuel flow is dependent upon three major factors: (1) the power output in excess of normal detonation limited output desired; (2) the importance of engine critical altitude performance to the operators; (3) the ambient or atmospheric temperatures in which the airplane is to operate.

Water alone offers the best rate of engine cooling. Water-methanol mixtures permit the greatest power output. Water-ethanol mixtures are inferior to either straight water or water-methanol.

The freezing points of methyl and ethyl alcohol solutions are shown in Fig. 2. Fortunately, the better antidetonant constituent – methyl – has the advantage, requiring approximately 30% less addition for the same freezing point.

B. Effect of Water Injection on Engine Brake Thermal Efficiency.

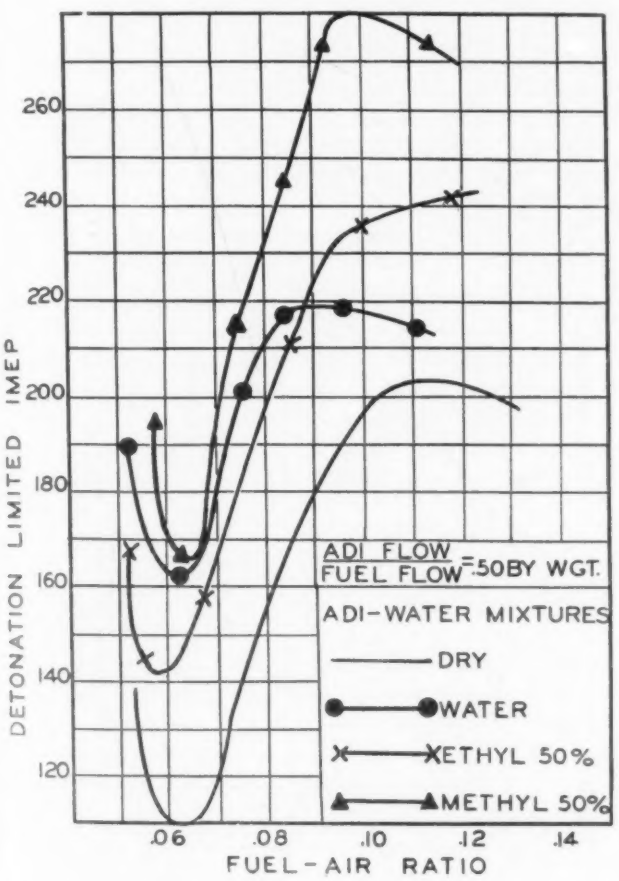
Of primary interest to the aircraft-engine operator, the effects of water injection on engine brake thermal efficiency are shown on constant power and speed mixture control curves of Figs. 3 and 4.



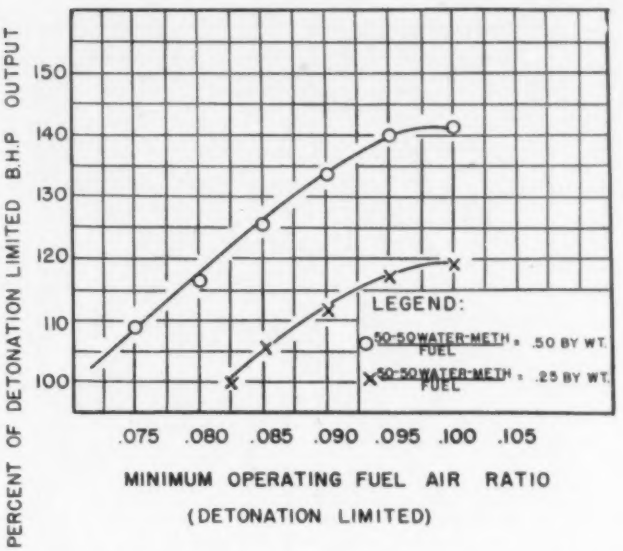
■ Fig. 5 – Typical effect of water injection on detonation limited imep – CFR engine

The data of Fig. 3 were obtained on a Wright Cyclone, 9-cylinder engine operated at low cruise power. These results are typical of cruise power operation from 30 to 60% of engine rated power and definitely indicate that it is not possible to obtain any increase in brake thermal efficiency by injecting water into the engine induction air. On the contrary, the injection of 100 lb per hr of water, approximately 0.17 lb per bhp-hr, caused a decrease in brake thermal efficiency of approximately 6%. This decrease is illustrated by the change in maximum economy or thermal efficiency shown by the hook portions of the curves of Fig. 3. The data are plotted against per cent best power fuel flow to emphasize the fact that best power fuel flow remains unchanged as water is added, and that engine specific air consumption is increased, causing a similar decrease in best power fuel-air ratio.

As engine power is increased above 70% of rated power, the hook portions of the curves defining maximum economy or thermal efficiency are brought together and there is no marked decrease in thermal efficiency. Fig. 4 is typical of power operation at 70% rated and above, and illustrates the change that has taken place. A water addition of 0.12 lb per bhp-hr, which previously caused a reduction in brake thermal efficiency of approximately 5%, apparently does not affect the thermal efficiency at this higher power output. Best power fuel flow, air specific, and fuel-air ratio behave the same at all powers.



■ Fig. 6 – Effect of various antidetonant fluids on detonation limited imep – CFR engine



■ Fig. 7 – Minimum fuel-air ratio requirements for increased brake horsepower output using water-methanol/fuel additions

C. Possible Increase in Detonation Limited Power Output.

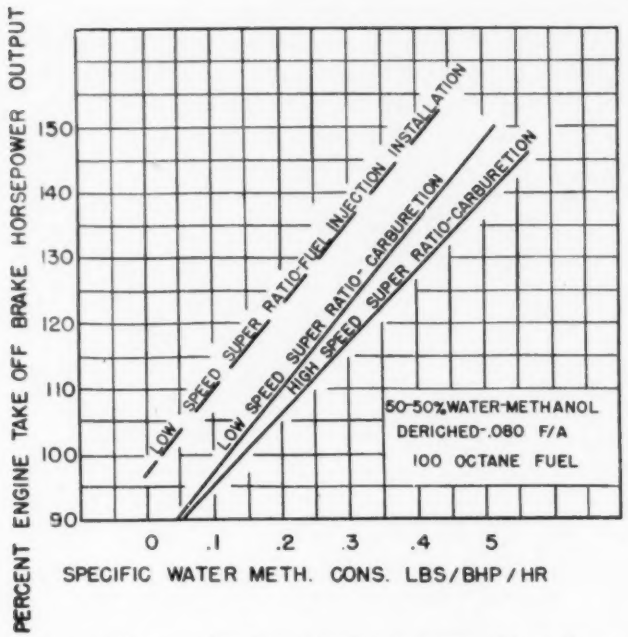
A major problem confronting aircraft-engine manufacturers during wartime is the provision of high output performance. Therefore, the greatest amount of water injection emphasis has been placed on power output in excess of engine take-off power.

Neglecting physical limitations of the engine proper, the maximum power output is dependent upon the amount of supercharging or final intake pressure available for cylinder intake. Since aircraft engines must provide high power output at altitude, there is always a reserve supply of supercharging available at any altitude below engine critical. The operating limitation generally preventing the use of powers in excess of normal rated or take-off is that of destructive detonation.

Figs. 5, 6, and 7 have been prepared to illustrate the effects of water, methyl, and ethyl alcohol mixtures on detonation limited indicated mean effective pressure. These data were run on a single-cylinder supercharged CFR engine for the purpose of establishing a working background for Wright Cyclone engines.

Although the general trend of these single-cylinder CFR engine curves is exhibited in full-scale engine operation, the percentage changes in detonation limited output do not completely agree and are, therefore, omitted from any discussion.

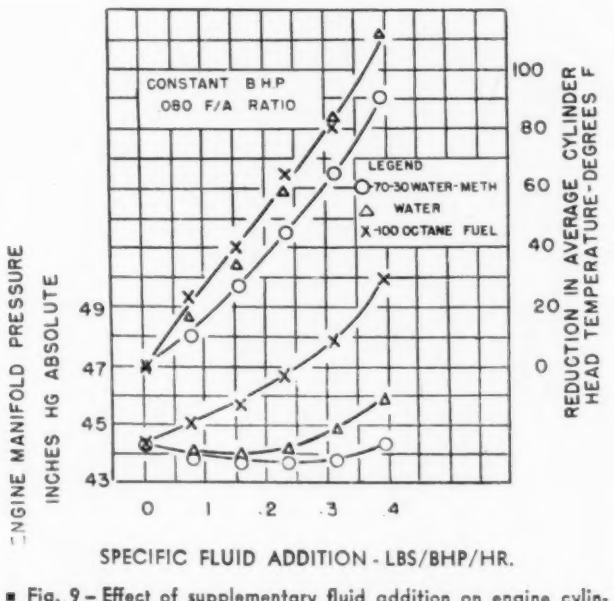
The data of Fig. 5 indicate that the addition of water to aircraft-engine fuel will permit a variable increase in detonation limited imep dependent upon the fuel-air ratio employed. These data also show that the percentage increase in detonation limited imep is reduced with an increase in water/fuel rate of injection. Full-scale operation has shown that water mixtures in excess of 50% by weight of fuel exhibit the tendency of drowning out the engine combustion cycle when operating at rich mixtures. This drowning tendency is best overcome by using water-alcohol mixtures instead of pure water and by operating at best power fuel-air ratios.



■ Fig. 8 – Water-methanol requirements for increased brake horsepower output at constant cylinder-head temperature – typical Wright Cyclone engine

The advantage of water-methyl alcohol mixtures as compared to pure water as a detonation inhibitor is shown in Fig. 6. Here, the addition of 50% methanol by volume to the water used has resulted in a considerable power increase. Although the ethyl-water mixture appears to be superior to straight water at rich fuel-air ratios, this conclusion is not borne out by full-scale testing.

Full-scale engine investigations of water injection made at the Wright Aeronautical Corp. have indicated that water-methanol mixtures are best suited as detonation inhibitors, with straight water second, and water-ethanol mixtures last.



■ Fig. 9 – Effect of supplementary fluid addition on engine cylinder-head temperature and manifold pressure – typical Wright Cyclone engine

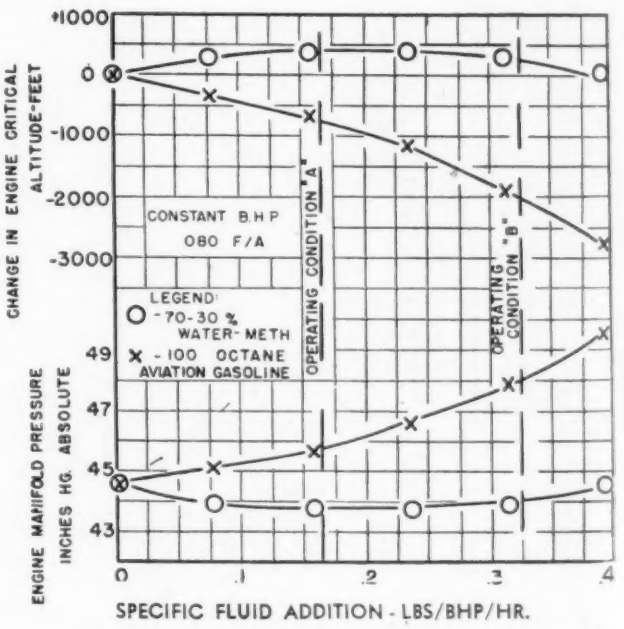
Fig. 7 has been prepared to illustrate water-methanol requirements for powers above normal, detonation limited power. The detonation limited power datum is based on operation of 100/130 grade aviation fuel at a 0.10 fuel-air ratio. The per cent of detonation limited bhp output is plotted versus fuel-air ratio, indicating the minimum fuel-air ratio which may be employed at any given power increase and antidetonant flow addition. The decreasing slope of power increase at rich fuel-air mixtures suggests the advantage of using lean mixtures and increased antidetonant flow, a conclusion borne out in actual full-scale practice.

D. Specific Coolant Requirements at Constant Engine Head Temperatures.

One of the most severe handicaps in operating aircraft engines at increased power output is the cooling penalty which will occur unless proper consideration is given to the quantity of antidetonant flow required per increase in power output. Any increase in engine head temperature above the manufacturer's maximum permissible means reduced engine life, shorter periods between major overhaul, increased operating expense. Increased cowl flap openings to maintain constant temperatures mean higher cooling drag, less critical altitude, increased cost per operated mile.

Fig. 8 illustrates antidetonant flow requirements for constant speed power increase at no increase in cylinder-head temperatures. The function of brake specific water-methanol consumption versus per cent engine take-off brake horsepower output is linear. Its slope is dependent upon the engine mechanical efficiency and therefore varies with supercharger speed ratio for any given engine. The data were obtained at 0.080 fuel-air ratio to afford maximum wide-open-throttle output at this approximate best power mixture.

A comparison of low blower operation for a carburetion



■ Fig. 10 – Effect of supplementary fluid addition on engine critical altitude – typical Wright Cyclone engine

and fuel injection installation for the same basic engine is shown in Fig. 8. Based on operating mixtures, the difference in specific water consumption requirements, at an equivalent power output, indicates that the fuel injection installation may be operated at 12% leaner mixtures than the carburetion installation for equivalent cooling. The basis of engine cooling comparison for water and fuel specifics will be illustrated at a later point in this paper.

E. Effect of Fuel and Water on Engine Cooling and Altitude Performance.

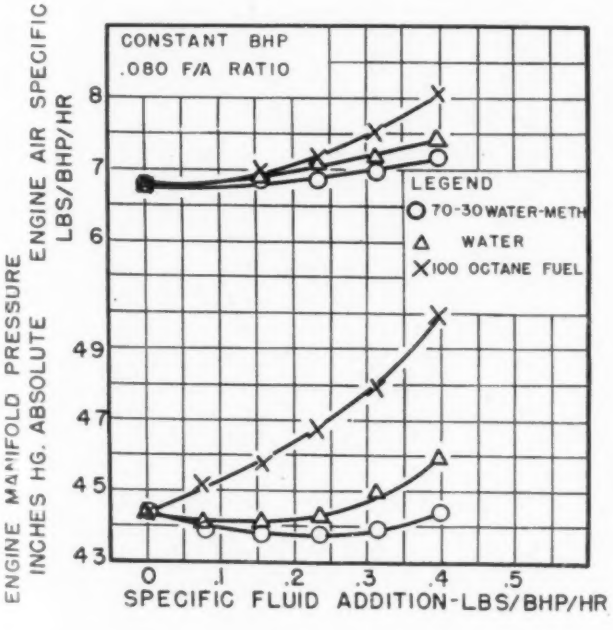
A conclusion which we wish to make at this point is that pure water is approximately equal to fuel when used as an engine internal coolant at high power output. This factor is important because it represents a possible fuel cost saving of approximately 25% with a small increase in fluid weight.

Also, approximately 1000-2000 ft of engine critical altitude can be gained at take-off power by use of water cooling in place of fuel cooling – a very important consideration for commercial carriers that must operate from airports of several different altitudes, and that must carefully consider all payload increase possibilities.

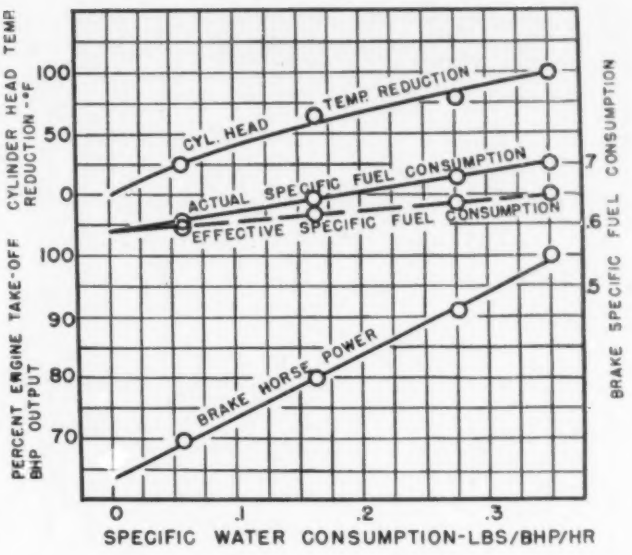
The effects of water, water-methanol, and gasoline injection as an engine internal coolant are shown in Fig. 9. The data were obtained at approximately 90% of engine take-off bhp output and a constant datum fuel-air ratio of 0.080.

The reduction in average cylinder-head temperature is approximately the same for fuel and water additions. Water-methanol mixtures are less efficient from the cooling standpoint – approximately 80% as efficient as water or fuel for the 70-30% water-methanol mixture used.

A disadvantage of the use of fuel as an internal coolant is shown by the 5 to 10% increase in engine manifold pressure requirements, representing a loss in engine critical



▪ Fig. 11 – Effect of supplementary fluid addition on engine specific air consumption – typical Wright Cyclone engine


CYLINDER HEAD TEMP REDUCTION -°F
CYL. HEAD TEMP REDUCTION
ACTUAL SPECIFIC FUEL CONSUMPTION
EFFECTIVE SPECIFIC FUEL CONSUMPTION
PERCENT ENGINE TAKE-OFF BHP OUTPUT
BRAKE HORSE POWER
BRAKE SPECIFIC FUEL CONSUMPTION
SPECIFIC WATER CONSUMPTION-LBS/BHP/HR

▪ Fig. 12 – Water consumption requirements for engine operation with 87-octane fuel – Wright Cyclone 9 engine

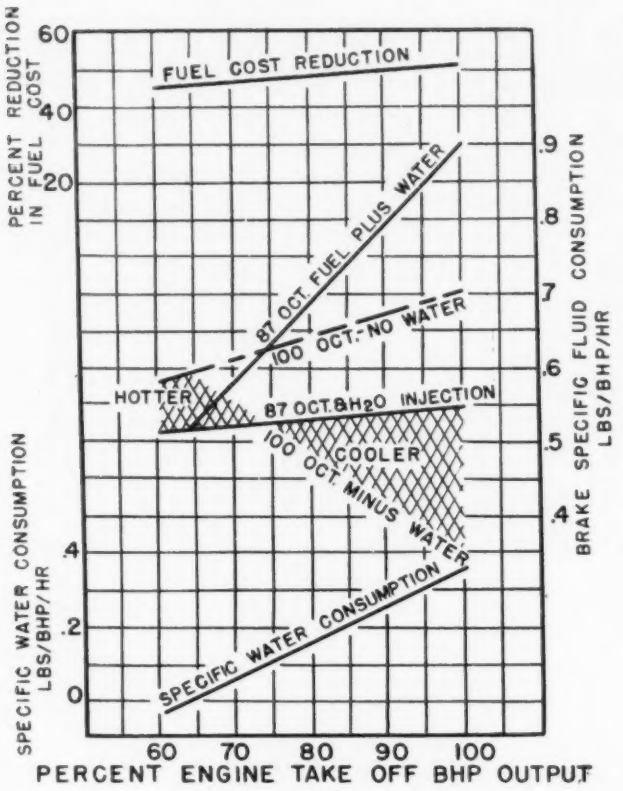
altitude. This loss, shown in Fig. 10, amounts to 1000-2000 ft, depending upon the operating condition desired.

As an illustration, operating condition *A*, shown in Fig. 10, represents a normal operating condition of 0.10 fuel-air based on fuel cooling addition. The curve data show a loss of approximately 1200 ft of critical altitude. Operating condition *B*, representing 120% of power *A* at the same 0.10 fuel-air ratio plus cooling fuel necessary to maintain constant temperatures, shows a critical altitude loss of approximately 2000 ft.

A review of Figs. 9 and 10 emphasizes the need for carefully considering the aircraft operational needs before selecting the antidetonant or cooling fluid to be used: water for maximum cooling, water-methanol for best detonation inhibitor and maximum altitude performance at high power operation.

Fig. 11 illustrates the effect of supplementary fluid addition on engine brake specific air consumption. Although air specific increases with fluid addition, the increase in engine volumetric efficiency apparently offsets the pumping losses of increased air consumption. The increase in volumetric efficiency is due to the intake charge temperature reduction obtained with water and water-methanol injection. The methanol mixture is most efficient and offsets pumping losses for a 0.4 specific fluid addition. The straight water is next best, offsetting losses for a 0.25 specific fluid addition. This factor, shown by the manifold pressure requirements, does not hold for gasoline because of its small heat of vaporization (150 Btu per lb) as compared to that of water and water-methanol mixtures (900-1100 Btu per lb) at the absolute intake charge pressures accompanying high power operation. The increased operating efficiency of the water-methanol addition as compared to pure water is believed due to a greater degree of vaporization during supercharging, and the subsequent increased cooling provided by the resultant change in variable specific heat during supercharging.

F. Water Injection with 87- and 91-Octane Fuels.



■ Fig. 13 – Engine fluid consumption analysis for operation with 87-octane fuel and water

In the past few years, the pace of war development has led to highly accelerated increases in bhp output per cubic inch of engine displacement.

The attractive resultant specific horsepower output and horsepower-engine weight ratios are not going to appear as attractive to the commercial operator. He, normally accustomed to the cost of 87- to 91-octane fuels, may suddenly find himself required to pay higher prices for the premium fuels necessary for our highly developed engines. He has the choice of accepting a lower specific power output, paying the premium cost of 100/130 grade aviation fuels, or using low cost fuel accompanied by water injection.

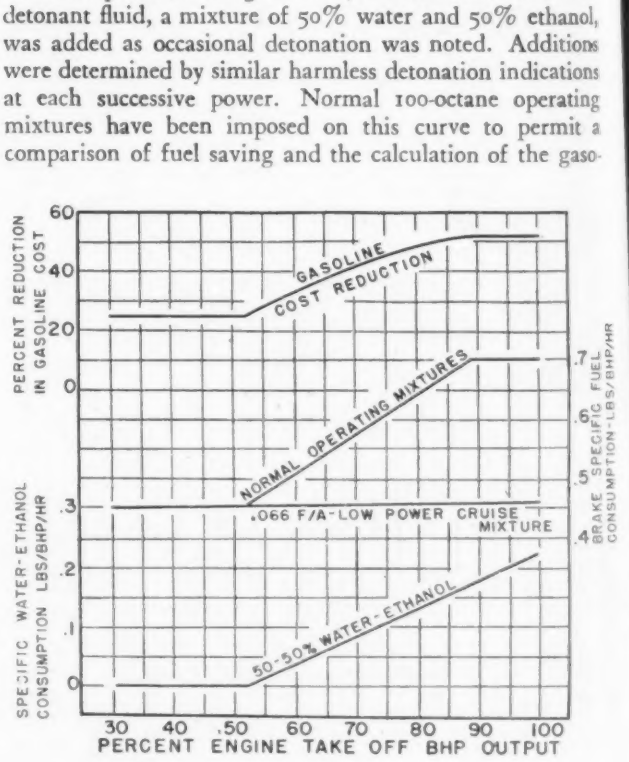
Figs. 12, 13, and 14 have been prepared to illustrate the possibilities of using high specific power output engines with 87- and 91-octane fuels.

The specific water consumption requirements for engine operation through take-off power are shown in Fig. 12. The actual specific fuel consumption shown is for both 100- and 87-octane fuels and represents normally used operating mixtures. The specific water consumption is the amount of water per engine brake horsepower-hour required to stay out of detonation. Note that the water addition has cooled the engine while preventing detonation. This cooling can be converted into a saving in cowl flap drag horsepower, and then into engine specific fuel consumption. The saving is shown in Fig. 12 as the difference between actual specific fuel consumption and effective specific fuel consumption.

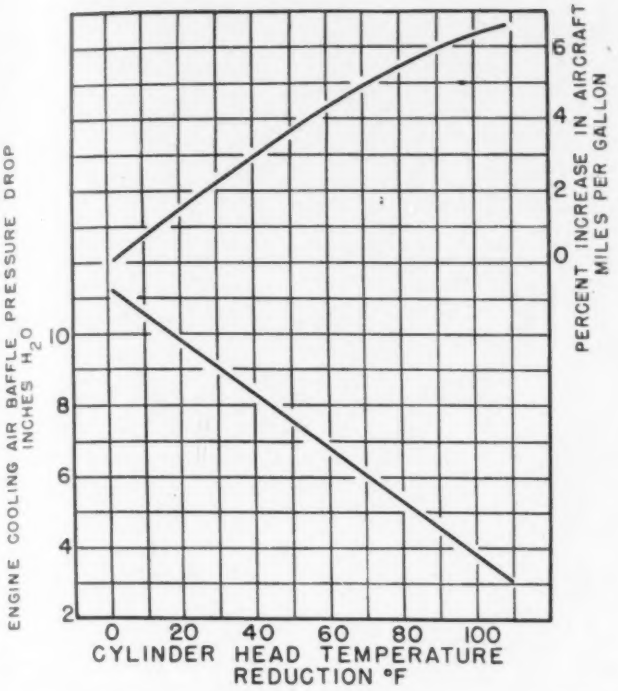
A better method of operating with the 87-octane fuel has been prepared in Fig. 13. Here, the fuel mixture has been deriched, taking advantage of the available increased cooling in a more direct manner. Actually, experience with water injection has shown that the lower octane fuels exhibit a greater tendency toward detonation at rich mixtures than at lean mixtures, probably because of the increased combustion-chamber flame speeds at the leaner mixtures. The difference in specific fuel consumption between the 100-octane and 87-octane fuels is 22% at take-off power instead of 7%, as previously noted for rich mixtures. Rating the water equal to fuel as an internal coolant permits us to draw the "100 octane minus water" line as the boundary of cooler and hotter portions of power output. Actually, a small revision of water flow at powers below 74% of take-off would permit operation of equal cylinder-head temperatures for both 100- and 87-octane fuel. The additional cooling above 74% power has been retained, since the water flow was determined necessary to prevent detonation. The overall weight penalty is shown as the brake specific fluid line, 87-octane fuel plus water. This penalty at 100% power is 28% fluid weight as compared to the 100-octane fuel and drops to zero at 75% take-off power. The additional fluid weight might easily be justified by the 51% saving in fuel cost shown by the per cent reduction in fuel cost curve.

Unfortunately, no 87-octane fuel studies have been made at low cruise powers, and therefore our 87-octane analysis must assume that automatic lean mixtures may be used with 87-octane fuel at low cruise powers. To lend credence to this statement, Fig. 14 has been prepared to show that 91-octane fuel has been used at low cruise powers and lean mixtures without the necessity of water addition.

The data of Fig. 14 were obtained at a fuel-air ratio of 0.066 for powers from 30 to 100% of take-off. The anti-detonant fluid, a mixture of 50% water and 50% ethanol, was added as occasional detonation was noted. Additions were determined by similar harmless detonation indications at each successive power. Normal 100-octane operating mixtures have been imposed on this curve to permit a comparison of fuel saving and the calculation of the gaso-



■ Fig. 14 – Engine fluid consumption analysis for operation with 91-octane fuel and water-ethanol – Wright Cyclone 9 engine


ENGINE COOLING AIR BAFFLE PRESSURE DROP INCHES H_2O
PERCENT INCREASE IN AIRCRAFT MILES PER GALLON
CYLINDER HEAD TEMPERATURE REDUCTION °F
0 20 40 60 80 100
2 4 6 8 10
0 2 4 6

■ Fig. 15 – Possible increase in aircraft miles per gallon of fuel obtained by reduction in cylinder-head temperatures

line cost reduction curve. The brake specific water-ethanol consumption is approximately 30% less than the water consumption requirements of the 87-octane fuel. This reduction is partly attributed to the difference in octane rating and partly because the 91-octane fuel data were run at lean mixtures. (Keep in mind the fact that the 87-octane requirements were determined at rich mixtures, even though the analysis of Fig. 13 was made at lean mixtures.) There is no fluid weight penalty attached to this 91-octane operation. Actually, a total fluid weight saving has been achieved throughout the high power range which varies from a 4.3% minimum to a 13% maximum. Although water-ethanol was used for this investigation, general experience has since shown that water-methanol or pure water is preferable. For this reason, the cost of the ethyl alcohol has not been taken into account when preparing the gasoline cost reduction curve. This curve once again illustrates the high saving in fuel cost which can be had by utilizing water injection – 52% at 100% power, decreasing to 25% at low cruise powers.

G. Effect of Cooling on Aircraft Operating Efficiency.

Having established the effective cooling possibilities of water injection, Fig. 15 has been prepared to illustrate the possible increase in aircraft miles per gallon of fuel obtained by reduction in cylinder-head temperatures. These data are based on cruise operation of a large commercial carrier traveling at an indicated air speed of 225 mph.

The effect of cooling drag change at constant thrust horsepower permits the calculation of decreased bhp requirements for constant speed flight, and therefore the decrease in gallons of fuel per mile, or conversely, the increase in miles per gallon. The relationship of cooling airflow, baffle pressure drop, and cylinder-head temperatures permitted plotting the data of Fig. 15.

Although these data are not generally applicable to any aircraft, their use with the water injection cooling data of preceding curves will permit a fair indication of the value of engine cooling to the aircraft operator.

H. Test Conclusions.

To sum up the most important results of the foregoing data:

1. Water provides the greatest degree of cooling at high power output.

2. Water-methanol mixtures are superior to water-ethanol mixtures.

3. Water-methanol mixtures provide the greatest increase in detonation limited horsepower output.

4. Water-methanol mixtures provide the greatest saving in engine critical altitude.

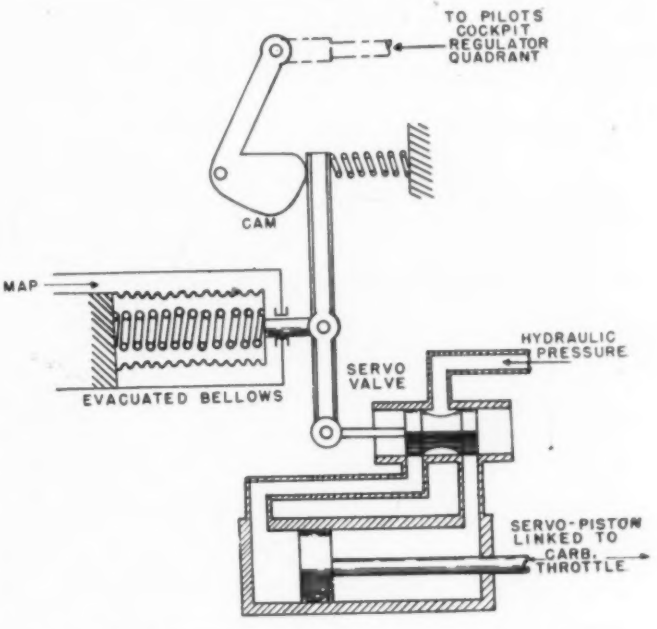
5. Pure alcohol additions should not be used for aircraft engines – they will cause a decrease in detonation limited output when using 100-octane fuel.

6. Water injection will permit the use of 91- and 87-octane fuels in place of 100-octane for equivalent power output up to take-off horsepower.

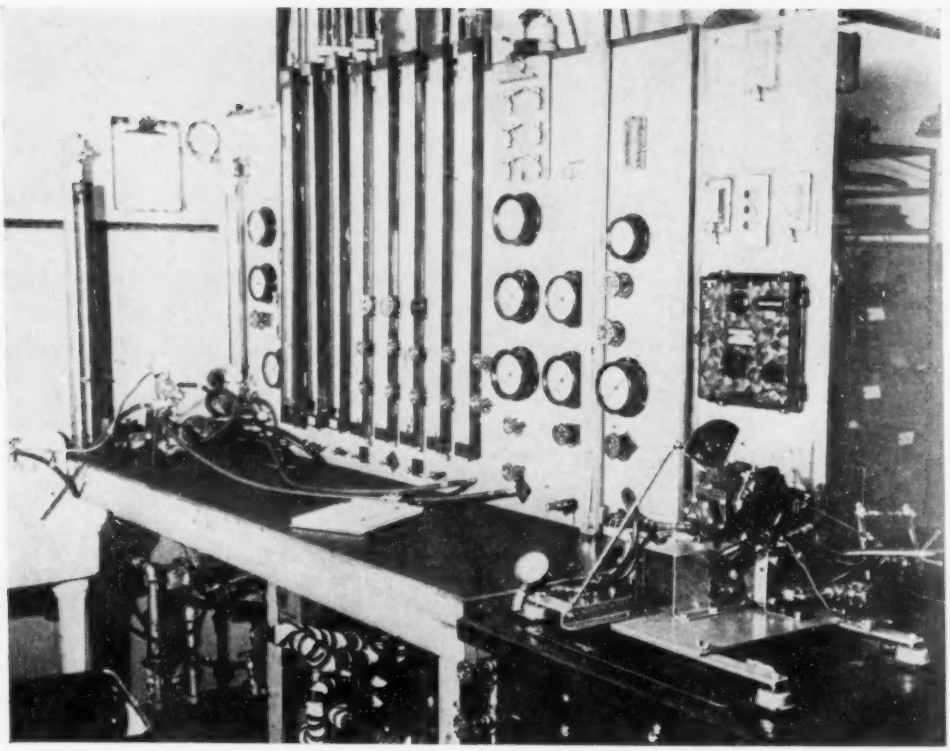
7. Water injection can be used most efficiently at lean fuel-air mixtures.

■ Automatic Injection Systems

A. Requirements – The problems of metering water and automatically controlling power are complex because of the many variables involved. A few of these are:



■ Fig. 16 – Schematic diagram of automatic manifold pressure regulator



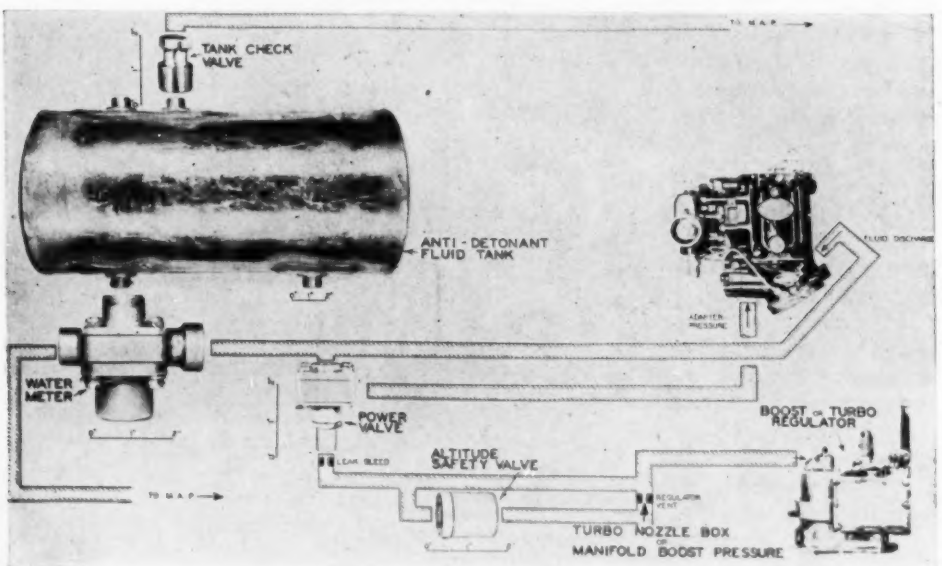
■ Fig. 17 – Test bench

1. Water distribution to the individual cylinders.
2. Effect of water on the fuel-air distribution.
3. Cylinder-head temperature variations.
4. Detonation limits.
5. Economy.
6. Engine roughness.
7. Engine corrosion.
8. Power.

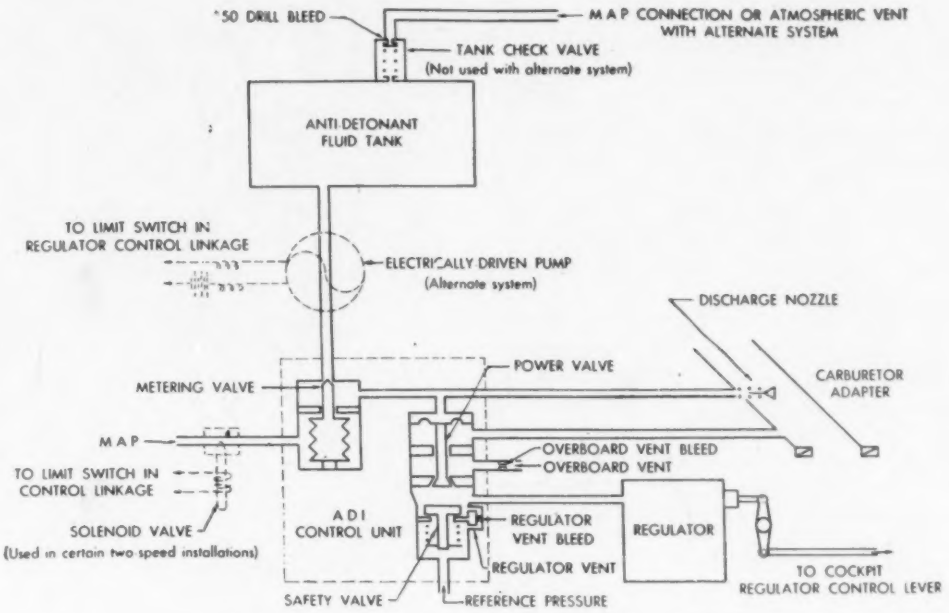
Distribution is a study by itself and, for obvious reasons, time is not available to expound on the theory and to review the work that has been completed. However, we have learned that relatively large tolerances can be used on methods and locations of injecting the antidetonant fluid into the engine and that the fuel injection engine does provide materially better distribution properties.

The problem of automatic water injection was fundamentally that of automatically regulating the engine manifold pressure to a prescribed value and to have this action accompanied by the proper flow of water into the engine. The requirements for a system as established were:

1. To prevent detonation.
2. The fluid must start flowing before the engine power is increased to the maximum high emergency rating.



■ Fig. 18 – Component parts of water injection system



■ Fig. 19 – Schematic diagram of water injection system

3. Engine power must be reduced automatically when the fluid supply is exhausted.

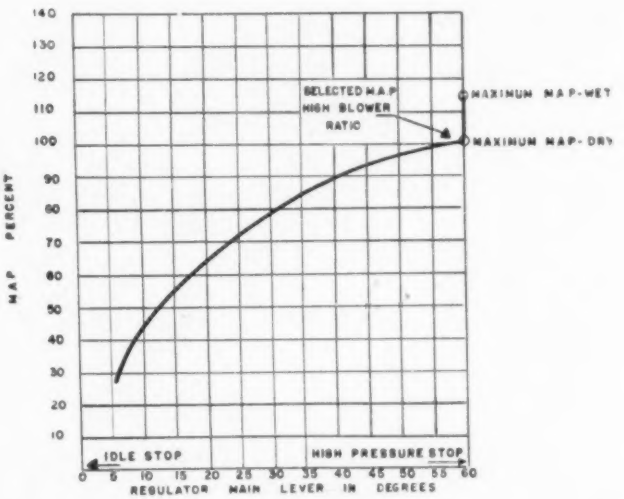
4. The system must prevent operation of the engine at ratings in excess of those normally limited by detonation unless the fluid is flowing.

5. The system must be generally applicable to any aircraft engine by use of setting variables change only.

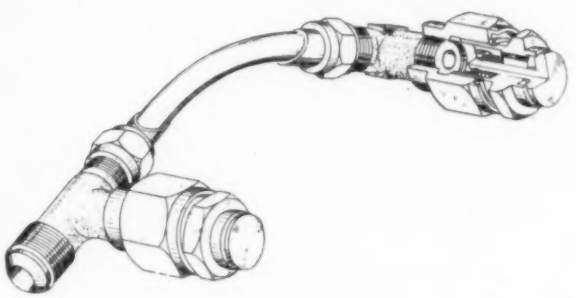
To make the system completely automatic, it is necessary to include an automatic manifold pressure regulator, engine power control, or the turbosupercharger regulator. The latter is used for aircraft turbosupercharger installations. The manifold pressure regulator controls engine manifold pressure by opening or closing the carburetor throttles. Manifold pressure is held constant at any desired value by use of a cockpit control lever. The regulator assembly is composed of a cam, bellows, servo valve, and piston arrangement which act to open or close the carburetor throttles in response to changes in engine manifold pressure. A schematic drawing of a typical manifold pressure regulator is shown in Fig. 16.

B. Test Equipment – In 1942 it was decided that Wright Aeronautical Corp. would undertake development of automatic water injection equipment for use with aeronautical engines. A laboratory was established specifically to carry on this development. It was used extensively to supplement engine testing and to study the function of the various components. A photograph of the test bench in the laboratory is shown in Fig. 17.

C. Component System Parts – Schematic illustrations depicting the original parts (fluid meter, power valve, and altitude safety valve) in series with a typical group of components (carburetor, water tank, and regulator) necessary to complete a system are Figs. 18 and 19.



■ Fig. 20 – Per cent increase in manifold pressure with corresponding movement increase in regulator main lever travel

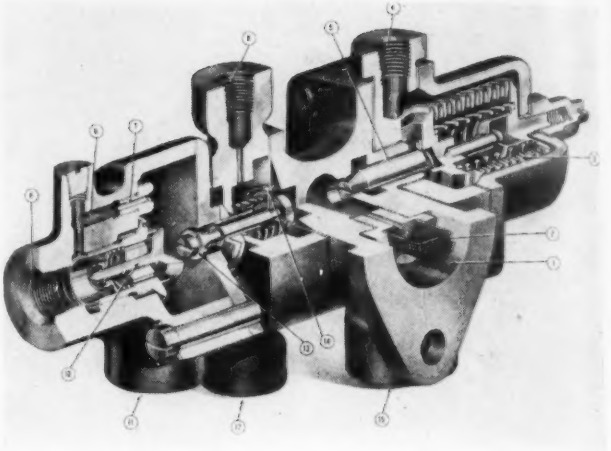


■ Fig. 21 – Cutaway drawing of simple, spring-loaded poppet valve discharge valve assembly as used in Cyclone engines

The operating cycle of the individual components is as follows:

1. The fluid meter controls the flow of antidetonant fluid by means of a needle valve; the latter controlled by an evacuated, calibrated bellows which in turn is sensitive to absolute manifold pressure. The regulator cockpit control lever generally incorporates stops to limit normal operating powers. At a point beyond the normal stop, enough manifold pressure is available to permit the automatic control system to operate. The bellows is calibrated to open the metering valve at a definite pressure condition and to permit pressurized antidetonant fluid to be injected into the engine via the carburetor adapter. The fluid is delivered to the fluid meter from either a manifold pressurized tank or a motor-driven water pump. The latter may be of the vane type, turbine type, or spur-gear type, and in construction closely resembles the well-known standard fuel pump. Noncorrosive materials and carbon vanes generally replace standard steel parts. Great pains are taken to eliminate electrolytic action due to the use of dissimilar metals.

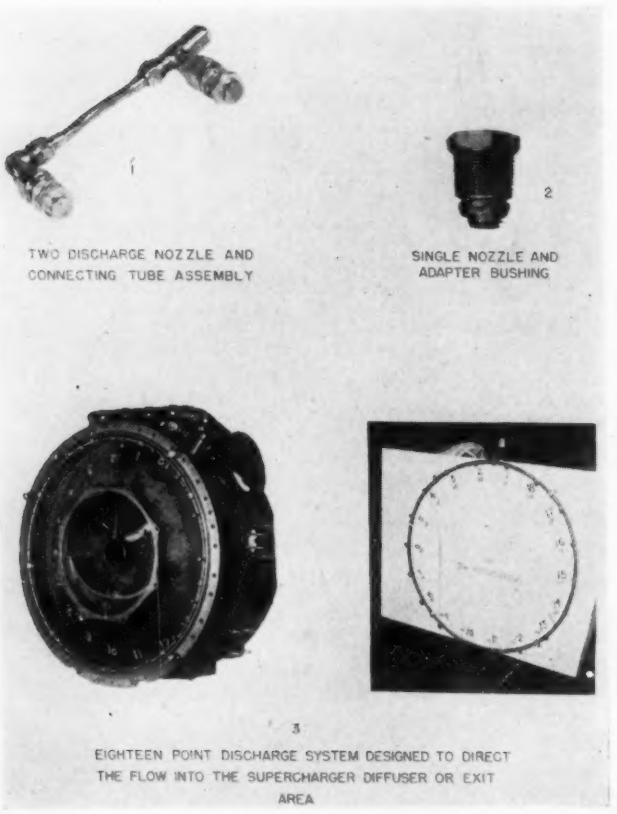
2. The power valve is controlled by antidetonant fluid discharge pressure and will not open its bleed passage until a flow of fluid is supplied. The bleed passage, when opened, vents off a controlled regulator pressure. The regulator, in turn, attempts to regain its lost pressure and builds up the pressure until the initial regulation pressure is regained. This automatically results in a definite increase in engine pressure and a consequent power increase.



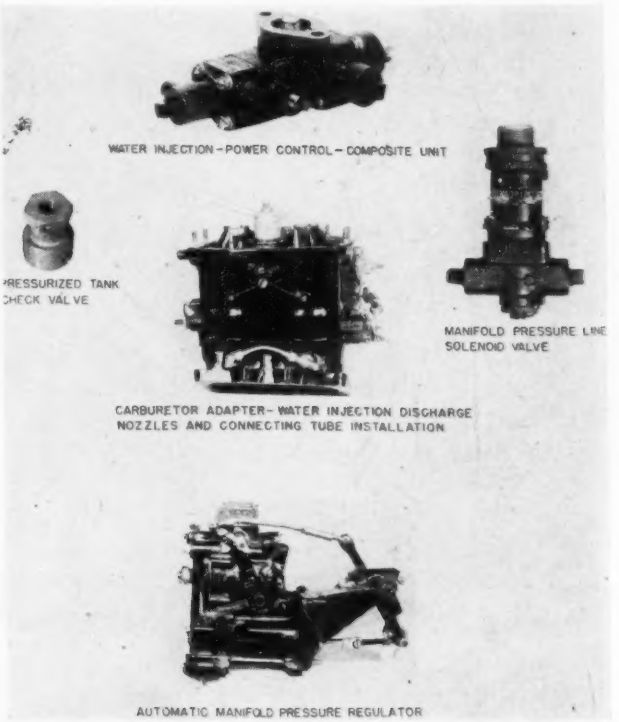
■ Fig. 23 – Cutaway drawing of ADI control unit

(1) Fluid in
(2) Strainer
(3) Evacuated bellows
(4) Manifold pressure in
(5) Needle valve
(6) Carburetor adapter vent
(7) Regulator vent restriction
(8) Regulator vent
(9) Reference pressure of boost pressure supply
(10) Safety valve
(11) Regulator pressure
(12) Overboard vent
(13) Power valve
(14) Power valve diaphragm
(15) Fluid out

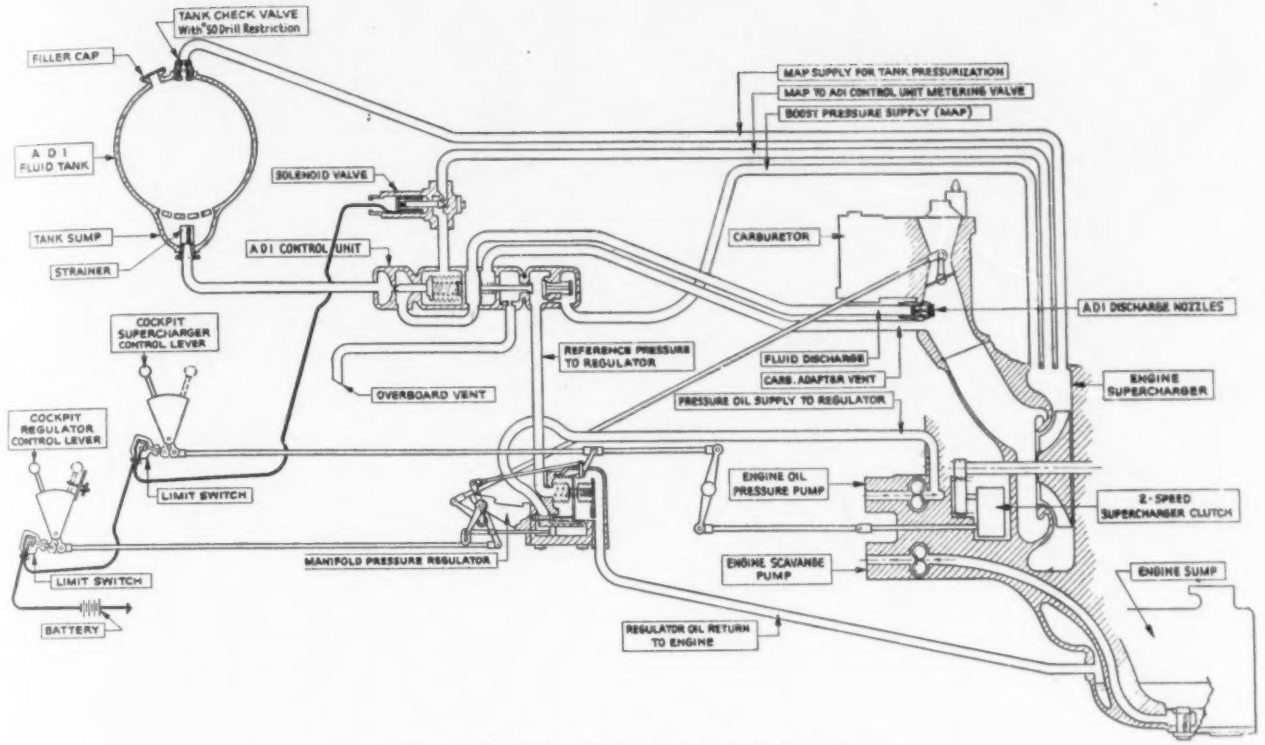
The per cent increase in manifold pressure with corresponding increment increase in regulator main lever travel is plotted in Fig. 20. The curve is representative of a Cyclone engine installation. By following the curve to its extremity, two manifold pressure values are available; the lower value represents the maximum pressure possible if no water is available and the higher indicates the per-


1
TWO DISCHARGE NOZZLE AND CONNECTING TUBE ASSEMBLY
2
SINGLE NOZZLE AND ADAPTER BUSHING
3
EIGHTEEN POINT DISCHARGE SYSTEM DESIGNED TO DIRECT THE FLOW INTO THE SUPERCHARGER DIFFUSER OR EXIT AREA

■ Fig. 22 – Water injection discharge systems


WATER INJECTION – POWER CONTROL – COMPOSITE UNIT
PRESSURIZED TANK CHECK VALVE
MANIFOLD PRESSURE LINE SOLENOID VALVE
CARBURETOR ADAPTER – WATER INJECTION DISCHARGE NOZZLES AND CONNECTING TUBE INSTALLATION
AUTOMATIC MANIFOLD PRESSURE REGULATOR

■ Fig. 24 – Water injection system – for use with manifold pressurized tank system



■ Fig. 25 – Schematic diagram of typical ADI system

missible pressure increase with water flowing into the engine. During normal flight operation (no fluid cooling requirement), when manifold pressure is not high enough to open the fluid meter valve, both sides of the power valve activating diaphragm are vented to the same pressure. Exhaustion of fluid and introduction of air to the fluid discharge line will result in a loss of pressure differential, closing the bleed passage, and automatically allowing the regulator to operate at normal manifold pressure values.

3. The safety valve is used to govern the regulator control drop for emergency powers. It is set for the peak pressure drop desired across the regulator vent and will limit the drop by bypassing regulator pressure from its original source into the regulator itself. This valve obviates the necessity of any complicated control of power bleed discharge pressure and does not allow changes in atmospheric pressure to vary emergency power pressure values.

4. Discharge valve systems: The method and means of injecting water into the aircraft engine are many. Some of the systems are as follows:

(a) Inject the antidetonant fluids with the fuel through the carburetor nozzle bar.

(b) Introduce the fluid into the centrifugal supercharger entrance with the aid of a spinner-type injector.

(c) Individual jet nozzles at the supercharger diffuser exit area.

(d) Individual cylinder injection, similar to fuel injection.

(e) Injection into the induction chamber below the throttle.

After an investigation to evaluate these methods, which included the evaluation of cylinder distribution, detonation, production and manufacturing, and service problems, it was decided to concentrate on the relatively simple poppet valve injection into the induction chamber for use with Wright engines.

A cross-section of a simple, spring-loaded poppet valve discharge valve assembly as used in Cyclone engines is shown in Fig. 21. One or two nozzles per installation are used, depending on the particular engine cylinder distribution requirements. Fig. 22 illustrates these systems. An experimental individual jet system, planned to inject fluid at the supercharger impeller exit, is pictured in the same figure. Although this latter system gave good distribution results, it had the definite disadvantage of possible discharge jet clogging, since the jet sizes were small.

The individual components of the water injection and power control system (water meter, power valve, and safety valve) were redesigned and combined into one small, compact, composite unit. The individual assemblies, already described, can be detected in the cut-away drawing of the Wright ADI control unit shown in Fig. 23. A photograph of this composite unit with the other parts furnished by the Wright Aeronautical Corp. for a complete water injection aircraft installation is shown in Fig. 24.

To illustrate further an aircraft installation, reference is made to the schematic diagram of a typical ADI system, as shown in Fig. 25.

concluded on page 44

Induction Ha

THE AUTHOR: F. F. VAUGHN has had over 25 years of metallurgical experience in the tractor and farm equipment industry, the past 16 of which have been with Caterpillar Tractor Co. – where he is now assistant chief metallurgist. For nine years prior to his present association, he was chief chemist-metallurgist for the Advance-Rumely Co.

by F. F. VAUGHN

Caterpillar Tractor Co.

THE principle of heat-treating by electrical induction is not new, yet it is only in recent years that the process has been used to any extent as a production method for various heat-treating operations. Rapid development in design has made available, today, modern equipment with split-second controls, so necessary in the successful use of the method in such operations. In fact, equipment is available that might rightly be classified as a series of machine tools for differential or localized surface hardening, and thorough or through hardening as well. Equipment capable of such precise control is essential for repetitive results and uniformity in the heat-treated parts.

■ Principle

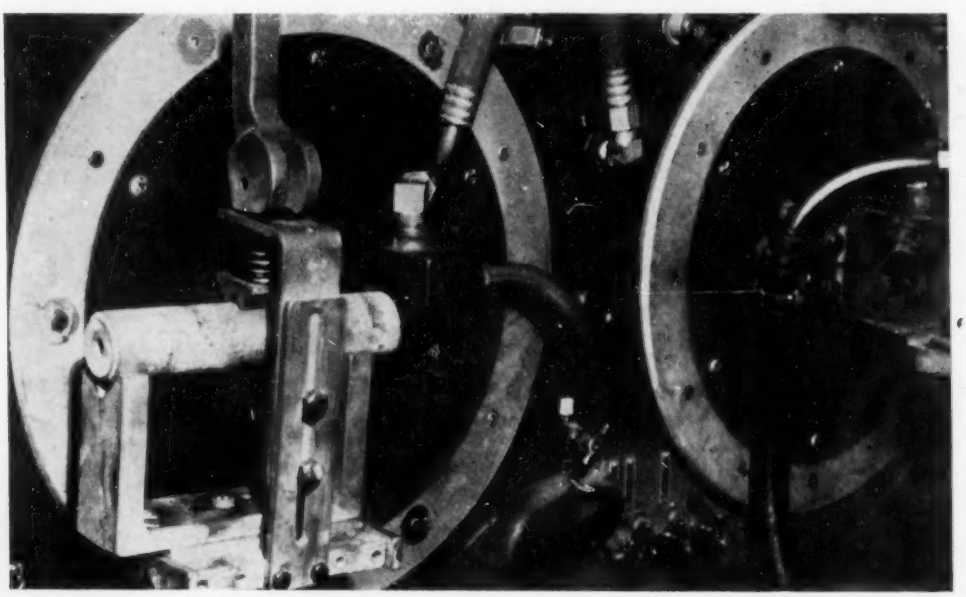
Induction heating is accomplished by converting electrical energy into heat, by arranging the work so that it is in the magnetic field created by high-frequency, alternating current through an induction coil. For example, if a steel part, such as a shaft, is placed within a wound coil or inductor carrying sufficient alternating current, as illustrated in Fig. 1, it becomes heated in that portion of the part within the magnetic field of its inductor. With enough power supplied, the heat developed will be sufficient to bring the localized surface zone of the part up to the critical temperature of the steel in a matter of seconds. Physical contact between the part and the electrical circuits is not required; hence, if desired, the heating may be accomplished progressively by moving the part.

With induction heating, most of the energy is changed to heat near the surface regions, and the higher the current frequency, the greater this tendency. This characteristic is known generally as skin-effect, and is an important consideration relative to the depth of hardening desired, particularly in surface hardening treatment. With increasing current frequency, the penetration of induced current (and the layer which can be directly and quickly heated) becomes shallower.

■ Surface Hardening

Because of the skin-effect characteristic, the method is ideal for surface hardening, hence the process and design of equipment has received greater emphasis in application to this type of heat-treatment. For such practice, the process requires relatively high power input for a time sufficiently *short* to minimize heat travel in the work by conduction. The ideal procedure requires sufficient power to confine the heating to just that depth to be hardened.

[This paper was presented at a meeting of the Milwaukee Section of the SAE, Milwaukee, Wis., Sept. 13, 1945.]



■ Fig. 1 – End of shaft in position in single-turn inductor – only portion of shaft that is within magnetic field of inductor is heated

Hardening as Applied to FARM MACHINERY

Obviously, to minimize heat travel by conduction, extremely short heating time is necessary. Such a heating cycle takes the affected zone to a high temperature, compared to conventional heat-treating temperatures, so that diffusion is very rapid, yet time at critical temperature must be so short that excessive grain coarsening, which would ordinarily be expected to accompany such temperature, does not occur. In fact rapid heating produces a better metallurgical structure in the hardened area, and a better bond between the hardened area and the core.

The extremely narrow limits within which power input and time must be controlled to produce satisfactory results is illustrated in Fig. 2.

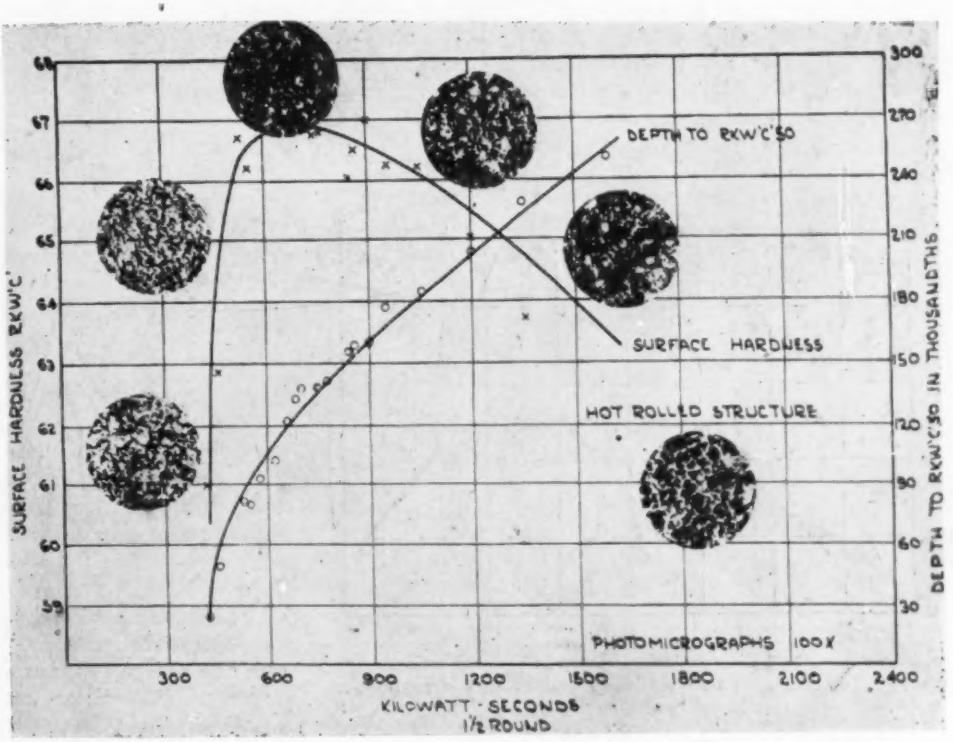
Excessive grain coarsening and resultant cracking along the grain boundaries, due to a few seconds longer time or slightly higher power input than necessary, is illustrated in Fig. 3.

Power input and time must, therefore, be under split-second control to ensure complete solution on the one hand, and prevent overheating and resultant disaster to the piece being hardened, on the other.

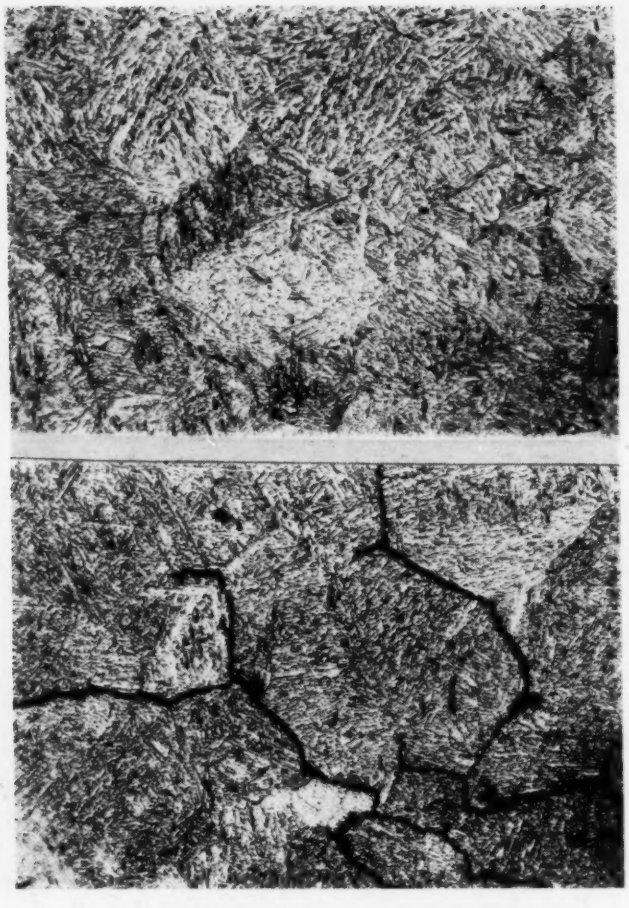
INDUCTION heating of a metal part is accomplished by placing the part in a magnetic field created by high-frequency, alternating current through an induction coil, thus converting electrical energy into heat and, with enough power supplied, heating the surface regions to the critical temperature in a few seconds.

Although this principle is not new, Mr. Vaughn points out that it is only in recent years that the process has been used to any extent as a production method for various heat-treating operations.

After mentioning some of the problems involved in using high-frequency current, the author gives examples of typical tractor and engine parts that are being successfully induction hardened in production.

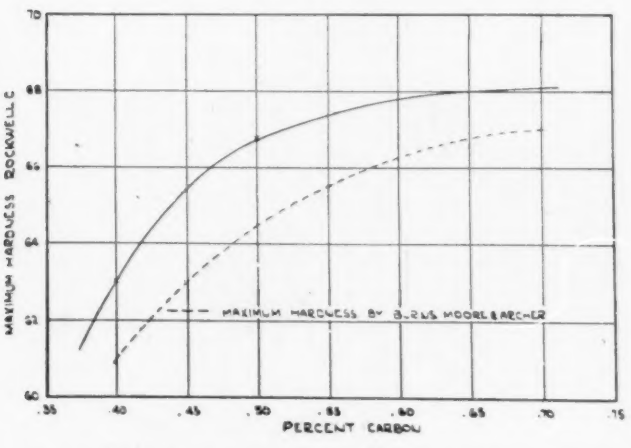


■ Fig. 2 – Chart illustrating extremely narrow power input time limits for satisfactory surface hardening



■ Fig. 3 – Grain coarsening (upper) and excessive grain coarsening (lower) with resultant cracking along grain boundaries

The method has been of special importance in conserving critical alloys by satisfactorily producing millions of surface hardened parts made of carbon steel. Adequate surface hardness can be obtained with fine-grain carbon steel of suitable carbon content. Maximum surface hardness is a function of carbon content, and consistent with proper metallographic structure, increases from Rockwell "C" 61 for 0.35% carbon steel, to a maximum of Rockwell "C" 68 for 0.70% carbon steel.



■ Fig. 4 – Maximum hardness plotted against carbon content
——— By induction hardening
- - - - - By conventional hardening

Fig. 4 shows maximum attainable surface hardness versus carbon content with both conventional and induction methods of heating. It will be noted that surface hardening by the induction method is capable of producing an appreciable increase in hardness above the maximum attainable by conventional methods.

We have successfully used high-frequency current for surface hardening millions of tractor and engine production parts of many types, such as crankshafts, track link pins, axle shafts, pump shafts, gears, and cylinder liners. Some of these parts require a considerable degree of overall toughness and strength for satisfactory service, hence, they are subjected to a preliminary heat-treatment in the rough or rough-machined state, prior to surface hardening. A major requirement of all these parts is high surface hardness to a specified depth for prolonging their service life. Such applications require that the parts be produced without overheating and resultant brittleness. Many of them were formerly carburized and hardened, others were made of alloy steel, directly hardened.

When proper heat-treating cycles are once established, they are capable of automatic repetition with extreme accuracy. We are producing these parts to a specified depth of hardness penetration within 10% plus or minus accuracy. In actual service, performance of the parts has been satisfactory.

On the following pages are depicted examples of tractor and engine parts that have been hardened by induction heating.

■ Full or Through Hardening

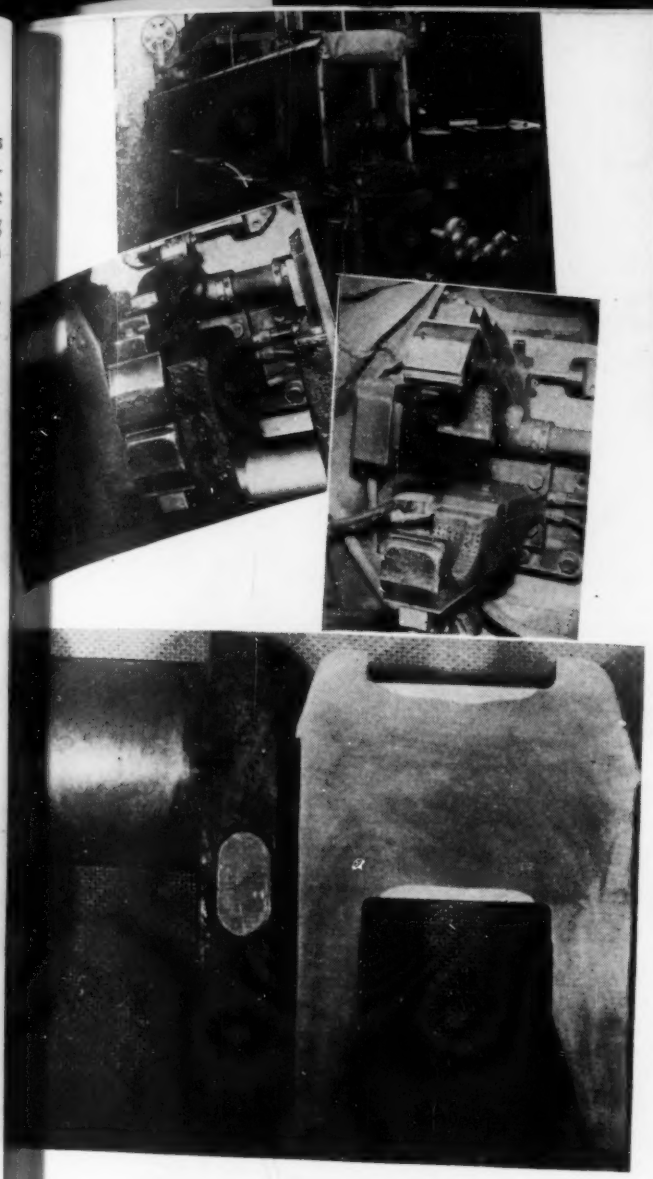
Recently, greater consideration has been given to adapting the induction heating method to the full, or through hardening type of heat-treatment used for enhancement of overall physical properties. In contrast to surface hardening, which requires relatively high power input for a time sufficiently short to *minimize* heat travel in the work by conduction, this type of heat-treatment requires a relatively lower power input for sufficient time to *allow* the heat to travel throughout the full cross-section of the material being heated.

An important, desirable feature of the induction method is the fact that heating can be accomplished without contacting the material being heated, thus, heating may be progressive by moving the material through the inductor coil, and in a protective atmosphere if desired. For example, with 9600-c current, using 30-kw power input, a 1-in. diameter bar may be heated to the hardening temperature throughout its cross-section while traveling through the coil at a rate of 22 in. per min.

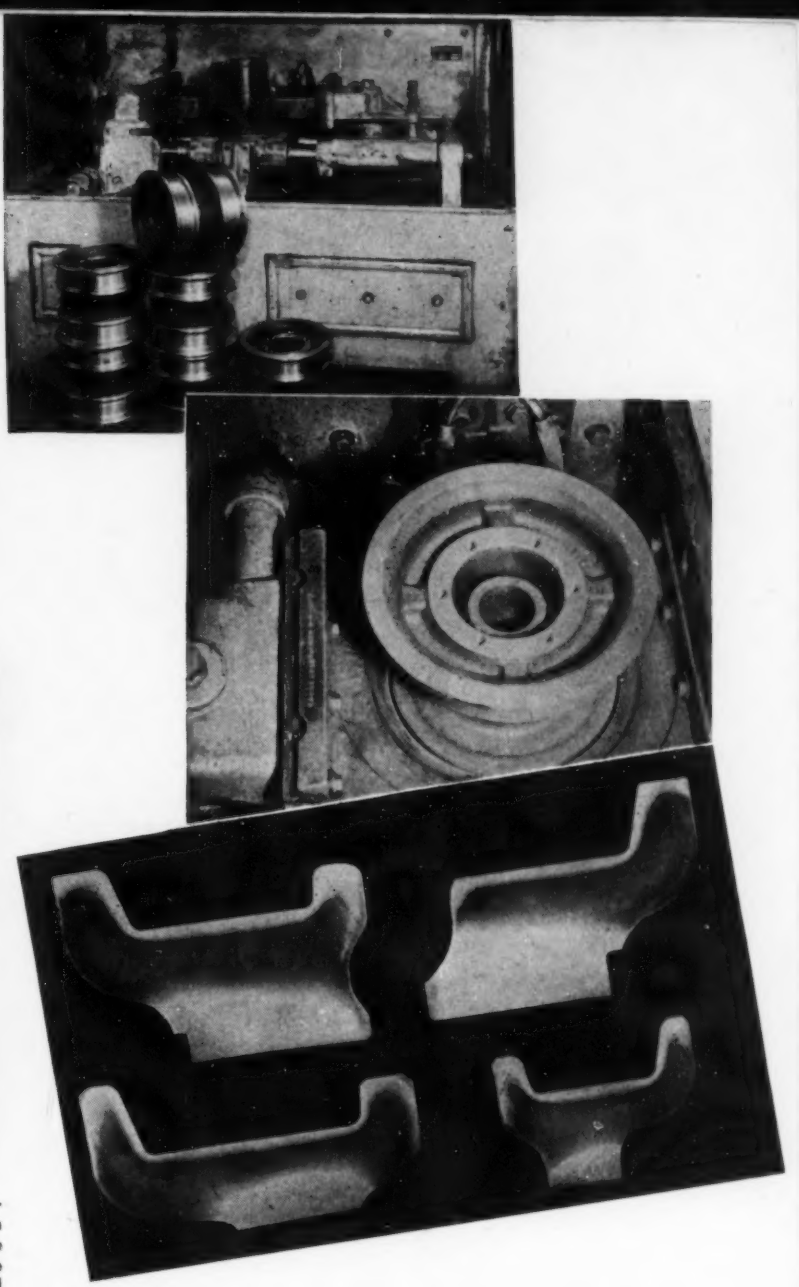
■ Application

A practical adaptation of this method has been in use for approximately a year for the continuous heat-treatment of cold-finished steel bars at the Caterpillar Tractor Co. Up to the present time, production has been limited to heat-treatment of 18-24 ft bars ranging from ½ to 1¼ in. in diameter.

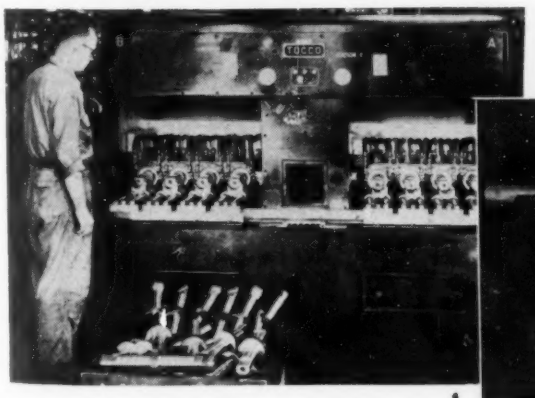
The bars are hardened and tempered progressively in a 4-station installation operating on 9600-c current supplied from two generating units, one for hardening and one for tempering. The stations may be operated simultaneously, or individually. When operated simultaneously, either the

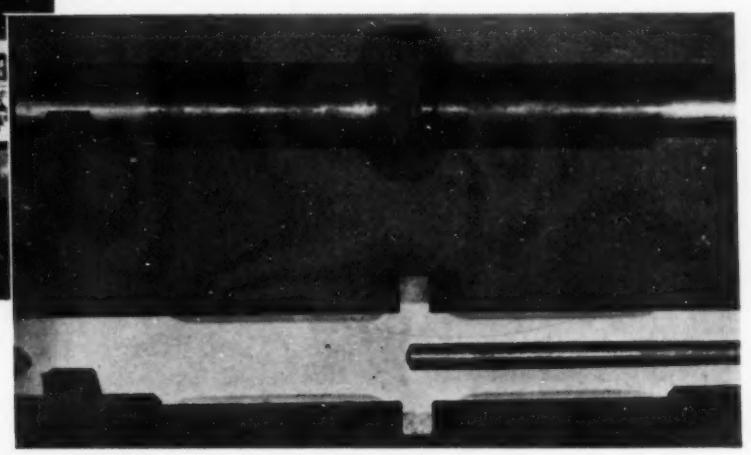


Bearing areas of crankshafts are surface hardened for greater resistance wear. An 8-station unit operating on 3000-c current from one common generating source is used for the purpose. The multistation tunnel line for hardening crankshafts is shown in the upper view. The center views show (left) the split inductor closed around a crankshaft bearing and (right) the split inductor in the open position. The lower view shows an etched cross-section of a hardened crankshaft bearing

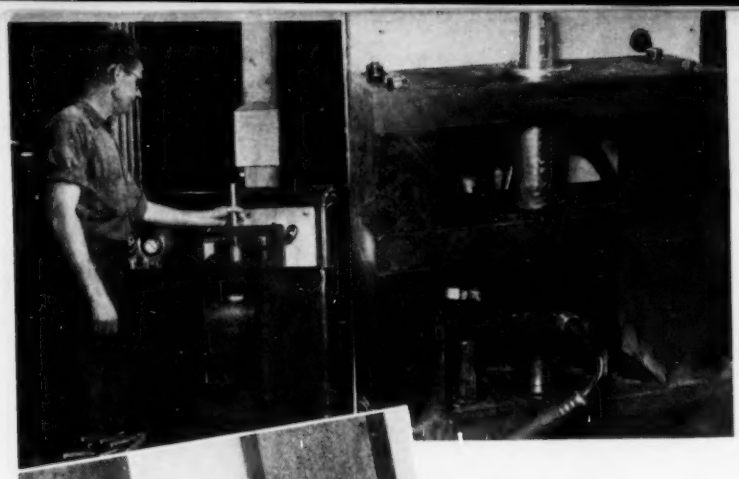


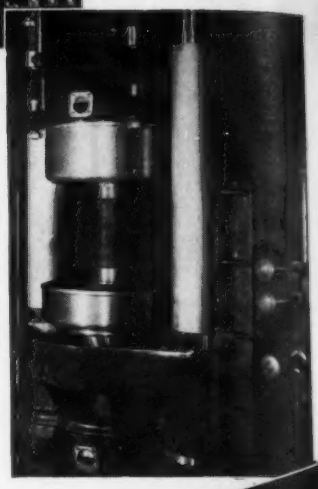
Track roller threads and flanges are surface hardened in several stations operating on 3000-c current from a common generating source. Track rollers and hardening station are illustrated in the upper view and the split inductor is shown closed around the thread of the track roller in the middle view. Cross-sections of typical hardened track rollers are shown in the lower view





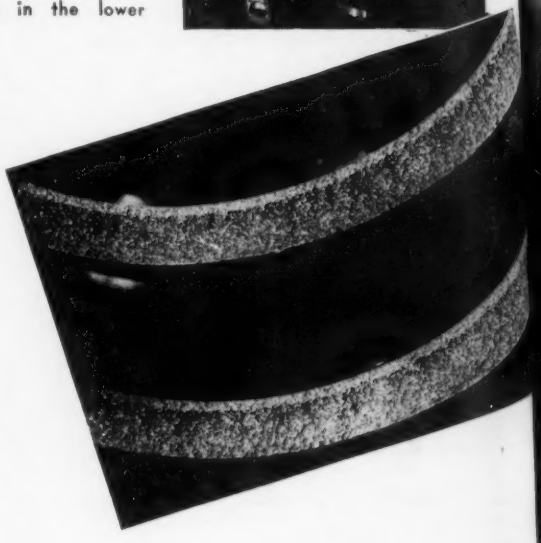
Track roller shafts are surface hardened in unit stations as shown in the upper view. Cross-section of a hardened track roller shaft is illustrated in the lower view.

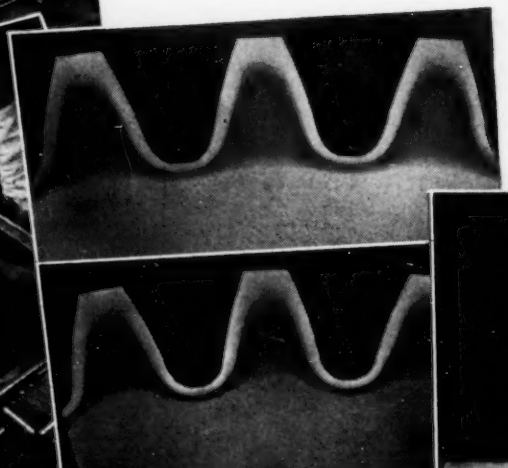


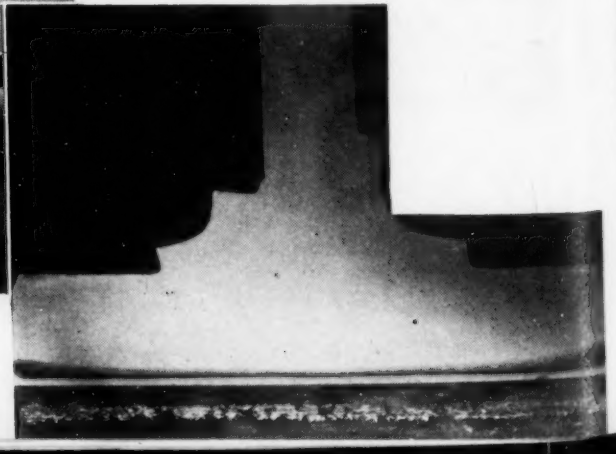


Cast-iron cylinder liners are surface hardened in the bore, using 3000-c current. The hardening operation is performed progressively as the liner is rotated. A machine for progressively hardening the cylinder inside diameter is shown in the upper view. A close-up of the cylinder liner, during the hardening operation is illustrated in the center view. Pieces of a cylinder liner, broken to expose the hardened bore, are shown in the lower view

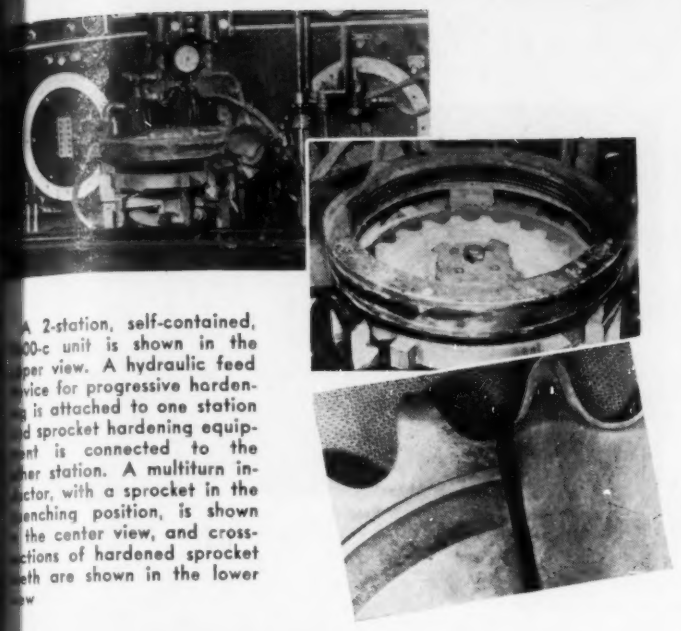
Track pins are progressively surface hardened in three stations operating on 3000-c current from one generating source. They are usually operated simultaneously, but they may be operated individually. A station for progressive hardening of track pins is shown in the upper view and a close-up of track pin inductor and guide tubes in the center views. A longitudinal cross-section of a hardened track pin is shown in the lower view



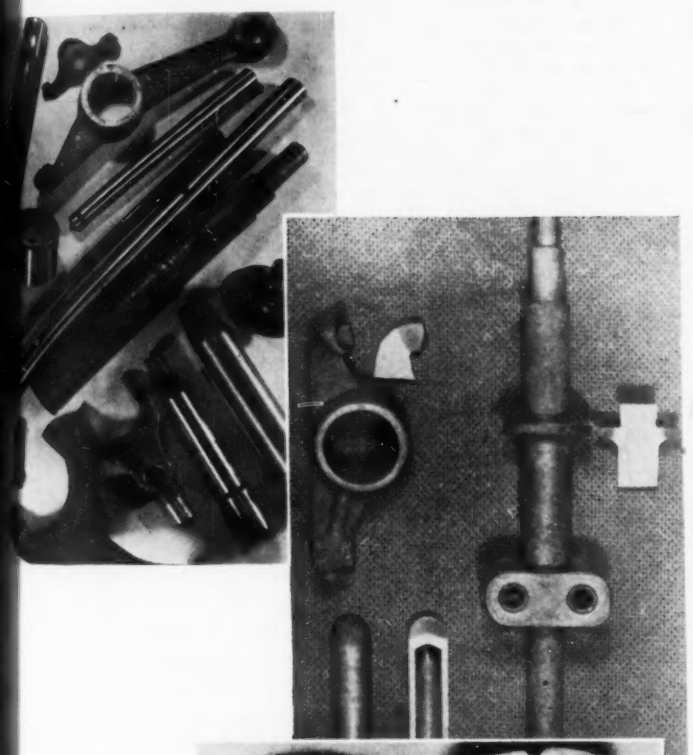




Final-drive gears up to 26 in. in diameter with a face width of 6 in. are surface hardened on three stations operating 9600-c current from one generating source. A final-drive gear in the loading and unloading position is shown in the upper view and in the quenching position in the center view. Cross-sections of typical, surface hardened, final-drive gear teeth are shown in the lower views



A 2-station, self-contained, 00-c unit is shown in the per view. A hydraulic feed vice for progressive harden- is attached to one station d sprocket hardening equip- nt is connected to the her station. A multiturn in- ctor, with a sprocket in the enching position, is shown the center view, and cross- ctions of hardened sprocket eth are shown in the lower w



A group of ypical miscella- eous parts hard- ned on 9600-c, elf - contained nits is shown in he upper view. Cross-sections of miscellaneous hardened parts re shown in the center and cross- ections of a hardened draw- bar pin in the ower view

same size stock, or four different sizes of stock may be treated. The equipment is shown in Figs. 5 and 6.

After heat-treatment, the bars have a blue or oxidized coloration, but are free from scale. Bar size, that is, diameter, increases less than 0.001 in. as a result of the treatment. Fine-grain, SAE 1035 type steel is used and the bars are produced to hardness Rockwell "C" 20-28, with an approximate yield strength of 110,000 psi and 80-100 Izod impact foot-pounds.

■ Advantages of Induction Hardening Process

The small floor space required, adaptability of the equipment, and the uniformity and speed of operations afford many advantages not obtainable with conventional equipment used for the heat-treatment of a great variety of parts. In many instances the following costly and time consuming operations usually associated with conventional methods of heat-treating may be eliminated or greatly minimized: Cleaning, distortion, straightening, decarburization, protection against carburizing, and material handling.

■ Conclusion

The many applications and examples illustrated and described could not have been produced on a production basis were it not for the precise electrical and mechanical controls provided in the induction equipment available today for heat-treating operations. However, these controls provide only a means of precise automatic repetition of a cycle of heat-treating operations which must be established on the basis of *proper metallurgical considerations*. It cannot be emphasized too strongly that anything less than this may easily prove disastrous in the end results.

DISCUSSION

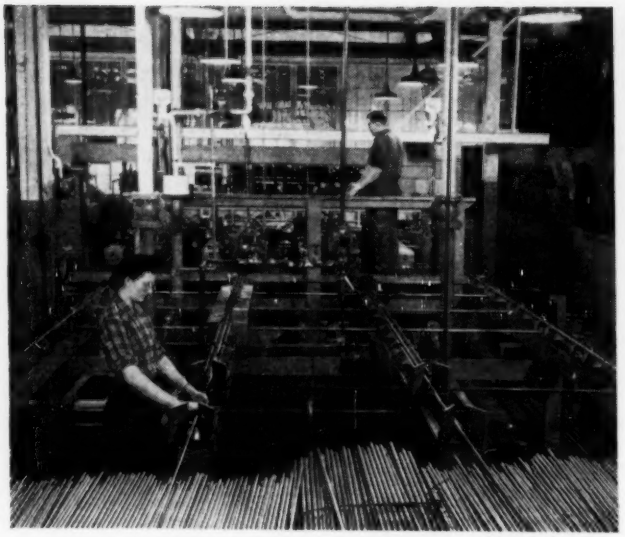
IT was generally agreed by the authors of the written discussion of Mr. Vaughn's paper that induction heating is here to stay and will find increasing application, despite the higher cost of the equipment. The greater precision possible, particularly with the recent developments in timing devices and the greater operating economies and the saving in time, compared with conventional methods of hardening were urged as reasons.

Certain precautions, however, must be taken, the discussers agreed. H. F. Kincaid, Farmall Works, International Harvester Co., Inc., for example, stressed the need for careful tooling and of rugged work-handling devices to permit repetition of established heat-treating cycles. Prior structure of the material is also important, according to Muir L. Frey, Allis-Chalmers Mfg. Co., who explained that for satisfactory metallurgical results, coarse or even moderately coarse structures are not acceptable. This indicates, he pointed out, that annealed structures usually are unsatisfactory, whereas hot-rolled or normalized structures are satisfactory in moderate-size sections.

Messrs. Frey and Kincaid were both particularly interested in Fig. 2, which shows the extremely narrow limits within which power input and time must be controlled to produce satisfactory results. They felt that more research should be done along these lines to determine how the curves would look if the frequency was varied considerably from that used in making the tests.

Mr. Frey discussed another important type of equipment not covered by Mr. Vaughn, namely, vacuum tube or electronic generator sets that operate at higher frequencies and are, therefore, capable of doing certain other types of operation. Its application varies from soldering operations to precise heat-treatments of steel in which the pattern and depth of hardening are closely controlled. Although this equipment is not yet productive of the high power outputs now obtainable from motor-generator type of equipment, the size of these vacuum tube sets is steadily increasing, and power outputs up to 100 kw are now commercially available. Available frequencies range from about 300 kc to about 15 megacycles, thus materially extending the application of induction hardening.

A technique of preheating before induction hardening, which increases the depth of any specified level of hardness, was also mentioned by Mr. Frey, who stated that preheating can be used in certain cases to control, within limits, the pattern of the hardened area.

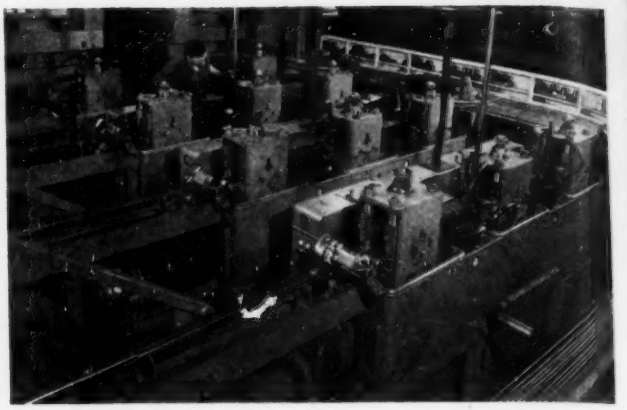


■ Fig. 5 – Entrance side of progressive heat-treating equipment of bar stock

H. Bornstein, Deere & Co., discussed the successful use of induction heating on a number of tractor items, such as tappet rods, power shafts, knuckles, and countershafts. In general, he stated, results on gears have not been satisfactory when using the motor-generator set or when using higher frequencies.

Successful use of induction heating for hardening the teeth of light-duty gears was then revealed by T. L. Burkland, John Deere Harvester Works, who described the use of a 9600-c induction heating unit used by his company, and explained that, in addition to its use in hardening gears, it was also used for hardening shafts to form the inner race for roller or needle bearings.

Length of shaft hardened varies from 1½ to about 3 in., and shaft diameter varies from ¾ to 1¼ in. Heating time is 3-8 sec and depth of hardened zone averages about ⅛ in. In this way shaft wear has been practically eliminated and, due to the greater load-carrying



■ Fig. 6 – View of bar stock hardening and tempering units

ability of the hardened shaft surface, it is possible to reduce the length of roller bearings used. This practice results in slightly lower bearing cost and reduced size of bearing and bearing box, thus allowing for some flexibility of design.

The light-duty gears are made of 1016 steel, the teeth being formed in forging and the gear being used as-forged, without any machining of the teeth. After forging, the gears are carburized, slow cooled, and shot blasted. The teeth are then heated by induction and finished, leaving the bore soft for machining. Mr. Burkland reported that this method raised heat-treating cost but has eliminated the distortion in the bore caused by heat-treatment, and has done away with the necessity of internal grinding to attain the desired fits; as a consequence, overall costs have been reduced.

Heating time of the gears varies from 8 to 25 sec, the entire tooth and root being heated. The carburized surface becomes file hard, the core hardness being normal for hardened 1016 steel. A rotating, air-operated lifting fixture is used in handling these gears on the induction machine. This fixture has four arms and a gear is placed on each arm. The gear is indexed under the inductor block and raised into position by the air-operated cylinder. After heating, the gear is lowered, indexed, and raised into a quench ring. While the heated gear is being quenched another gear is being heated.

Water Injection for Aircraft Engines

continued from page 37

D. Derichment Operation – As pointed out, by operating aircraft engines for extended periods of time at powers in excess of normal cruising, a resultant saving in fuel consumption is gained; consequently, the subject of fuel derichment supplemented with water injection cooling continues to take on added importance.

The requirements for a system were established as follows:

1. The derichment device on the carburetor must be completely automatic.
2. The fuel metering curve should be deriched to the basic metering curve or some intermediate point for the complete range of flows.
3. Water metering must be completely automatic for various powers.

E. Engine Corrosion – This problem has not been fully evaluated in aircraft engines; however, indications are that water injection has no deleterious effects. The parts of one Cyclone engine were examined carefully after 300 hr of water injection operation and no defects attributed to water corrosion were detected. Oil sample analyses taken of oil systems used with water injection engines gave no evidence of dilution. One sample of oil from an engine that completed 80 hr of water injection operation had a neutralization number of 0.4.

F. Newer Requirements and Conclusions – The progress in the past few years, brought about by understanding the variables involved in water injection application, has been noticeable in the aircraft industry. The use of water injection as a detonation limiting medium, has made remarkable increases in engine power ratings possible. Operation at these high powers is critical and, consequently, close control must be maintained of the many variables involved. In line with the latest requirements, water injection systems are being developed which give automatic water injection with selective power operation over the complete high power range of engine operation.

This discussion has been limited to the main factors that are necessary to cover water injection theory, testing, equipment, and its application to aircraft engines. There are detailed phases of antidetonant injection that are closely related to our discussion but are beyond the scope of the general treatment discussed herein.

The part played by water injection in military aircraft applications has been established and, undoubtedly, development will be continued. It is visualized that commercial operators will analyze the presented information with the purpose of directing it into their plan of operations. Their participation will be dependent on postwar aspects.

INDUCTION HEAT-TREATMENT of Internal Surfaces as Applied to Automotive Industries

by HOWARD E. SOMES*

THE use of induction heat-treating, combined with precision quenching, particularly where only a portion of a product requires selective treatment, was one of the major applications of new processes which made possible the use of low-alloy steels and cast iron during the war-time shortage of high-quality materials, according to Mr. Somes.

The application of the process to problems of the automotive industry is discussed by Mr. Somes, who suggests a number of products to which its use can be extended in the developments of the postwar period.

The author also recounts the experiments which went on in the development of the process and its application to the selective heat-treatment, by induction heating, of the Ford passenger-car hub, to provide by bore hardening an integral bearing race. In the course of this presentation, he details the development of the process and its present applications both in the use of static and progressive cylinder heat-treat machines.

■

THE AUTHOR: HOWARD E. SOMES (M '38), until recently chief engineer of Budd Induction Heating, Inc., is now semi-retired – after seven years' association with the Budd organization. Gaining his initial experience in the electrical end of the steel business at Algoma Steel Co., Mr. Somes gradually centered his interests in this field. He was with the Automatic Transportation Co. as chief engineer, with Chevrolet Motor Co. as electrical engineer, and later he formed his own engineering firm, Miller-Somes, Inc., to engage in engineering design and construction of industrial buildings. Before joining Budd, Mr. Somes was with the Detroit Edison Co. in a consulting capacity, his principal work being to find new sources for large-scale use of electric energy.

WHILE listening to the radio and reading the newspapers the past several years, most of us have been impressed with the fact that total war, involving as it does all the earth and its people, has posed new problems almost daily. Not only the military authorities, on whom the direction of our air, land, and sea forces depend, but most important, our scientists, engineers, and production people feel this very strongly.

■ Problems and Materials

For one thing, the use of materials commonly available before the war has been denied us or the supplies available greatly restricted. Many of these were alloys used to improve the physical properties or the performance and behavior of steels in manufacture and heat-treatment. In the development of better guns, better tanks, better aircraft and ships, in fact in the development of all equipment used for this new mobile war, machines of high output have been required, providing for higher striking power, greater speed, and greater endurance. To satisfy these needs, materials have been worked to a much higher level of strength, and we have needed at the same time to create designs which were light yet which would withstand the racking abuse of high speeds under the difficult conditions of dust, heat, cold, tropical rain, and the ice of the northern zones. Certainly, in no previous period have the engineer, the metallurgist, and the shop been required to develop so many new precise designs from such limited materials.

To achieve such great progress has required the breaking down of many obstacles. One of these great obstacles has been the natural tendency of man toward mental inertia – to accept new ideas slowly. However, necessity is a stern mistress and the mother of invention.

Throughout the history of the automobile and the development of the rubber age, rubber has been transported countless thousands of miles through the tropics and over the ocean to satisfy our expanding needs. All of us remember the academic interest with which we viewed only a few years ago the possibility of producing rubber from alcohol and petroleum as discussed in our technical press and, spasmodically, in our newspapers. One need look back only a year or two through the files of our various journals to recall the conflict of opinion as to whether synthetic rubber might ever be made with profit and still compete with natural rubber. Yet who today

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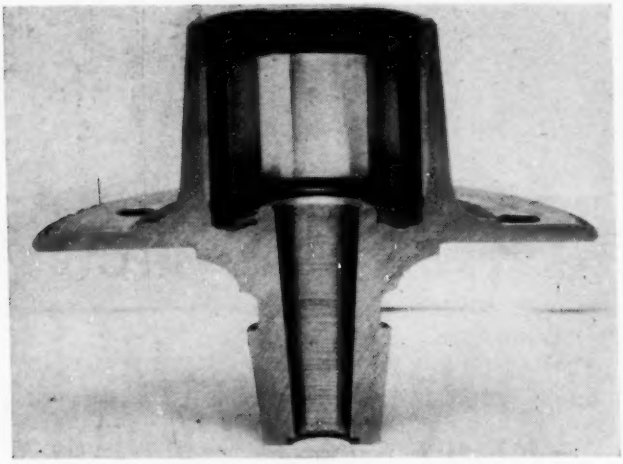


■ Fig. 1 – View of Ford hub

would venture to say that synthetic rubber may not ultimately become our principal source of rubber after this war is won. Here we have a great technological advance spurred through stern necessity, and now men best able to judge are convinced that synthetic rubber will be produced sufficiently competitively with natural rubber so as to be no great economic burden on the American people in the future.

Particularly in the field of metallurgy, we have experienced a serious reduction in the specifications of steels and critical alloys available; and to combat this we have witnessed applications of such new processes as the centrifugal method of casting steel so as to compete successfully with and often to excel forgings.

Probably there has been no field in metallurgy subject to more important and vital changes than the art of heat-treating. One branch of this art which has made phenomenal strides in the past few years is that embodying the application of electromagnetic induction as a means of heating, together with precision quenching, this method being particularly useful where only a portion of the product requires selective heat-treatment. The principle of heat-treating by electromagnetic induction is not new. Many scientists and inventors throughout the past 30 years have done prodigious work on many applications of the process. From the earliest conception of the importance of heat-treating, it was seen that electrical currents could be caused to flow through a selected path within a metallic body and that these currents would generate heat. It was but one step further to realize that currents of higher frequency would penetrate to a lesser depth.

Dr. E. F. Northrup was one of the first, if not the first, to realize the importance of high-frequency induction. His many investigations and the mass of data which he accumulated from 1915 to his death in 1940 have proved of great value to the many who were seeking to perfect methods of induction heat-treating of metals.

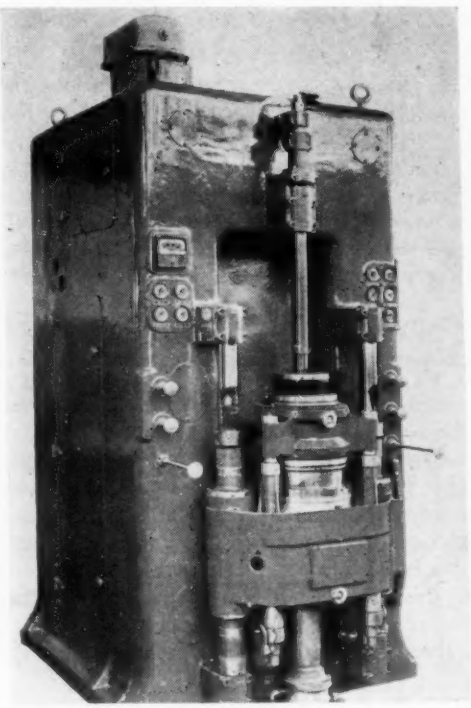
W. C. Dunn, president, and F. S. Denneen, treasurer and general manager, of the Ohio Crankshaft Co., were the first to put into large-scale use (early in the 1930's) the knowledge that induction heating, coupled with precise methods of quenching, could, if necessary equipment were available, greatly improve the bearing surfaces of crankshafts, camshafts, and external cylindrical surfaces. The Ohio Crankshaft Co. provided this vehicle, through which their experiments were conducted. The reader is undoubtedly acquainted with the successful work which has been done by this company in this field.

My own active association with the induction heat-treating began in 1934 when I was assigned by the Detroit Edison Co. to the task of developing new uses for electric energy. At the outset of this association it was natural that we should begin a search for new processes which could be made available to the automotive industry and which would involve the use of electric energy on a large scale. Among other fields studied was that of electric heating.

During this period, W. H. Farr, vice-president of the Budd Wheel Co., consulted with me on the problem of locally normalizing an automobile hub. It was desired that this hub have an integral bearing race, and to accomplish this it was being heat-treated in a lead pot. But it was found that once this hub was normalized, the stem section of the hub became too soft for good machinability when the barrel section was given the right treatment for good hardenability. I proposed to normalize this hub using 60-c induction in the barrel section. Several thousand hubs were so normalized. The next logical step was to think of ways and means of then selectively hardening a bearing race within this barrel.

Investigation followed and it was found there was no record of ever having successfully heat-treated the bore of a cylinder by electromagnetic induction. Invention followed invention, and a year and a half later this automobile hub, which was used on the Ford passenger car, was being successfully bore hardened to provide an integral bearing race. A number of millions of these hubs have been in most satisfactory operation for the past seven years. (See Fig. 1.)

Once we had demonstrated that the process could be used successfully, it became necessary to develop a produc-



■ Fig. 2 – Static heat-treat machine

tion line technique so that it could be fitted into a production cycle without slowing up the flow of work.

We know that a piece of metal when placed in a high-frequency magnetic field of sufficient intensity will heat to any desired temperature.

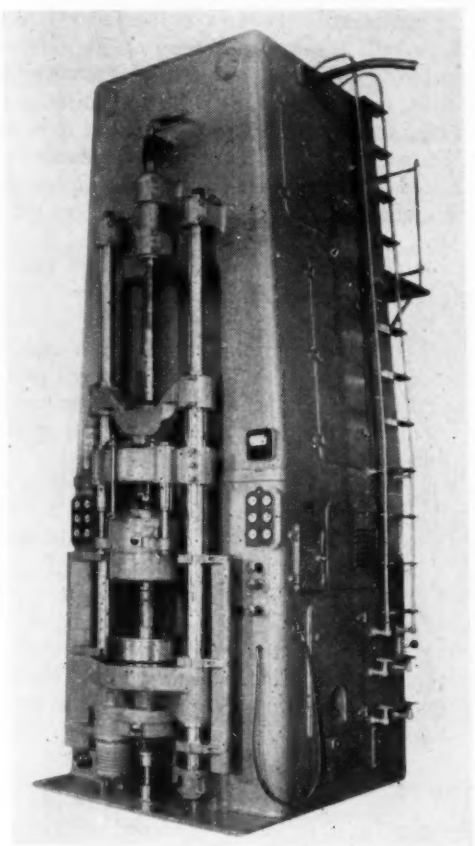
We know that the time of heating is controlled by the rate at which electric energy may be applied and the thermal constants of the material.

Most of us also have heard – probably at considerable length – of skin effect and that the higher the frequency of the magnetic field, the more marked the tendency to cause the heating currents to flow in a shallow path.

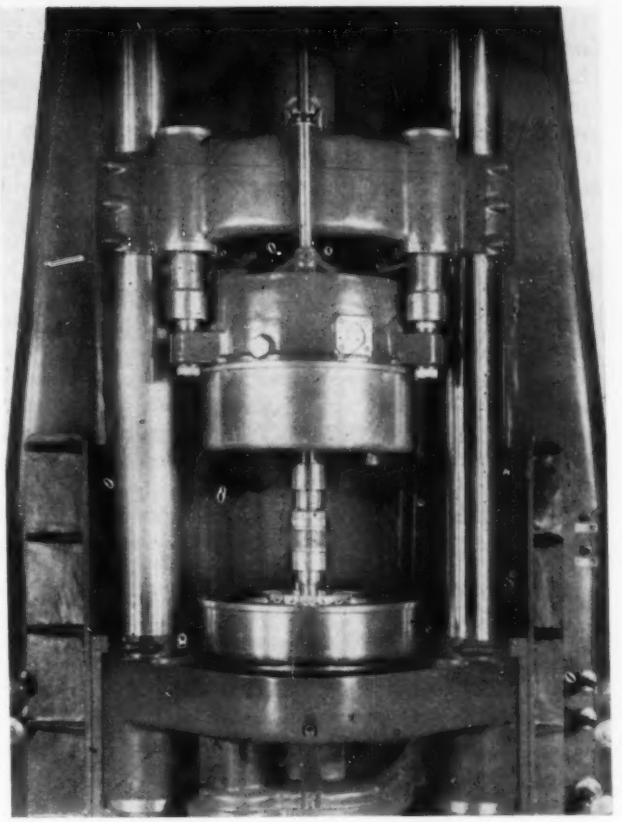
When induction heating is applied to a work piece at a rate exceeding, to a marked degree, the rate at which it will flow through the piece, we have local or differential heating.

This localized or differential heating has been very largely used for the heat-treatment of metal parts and usually takes place at an advanced stage of manufacture. In order to achieve this localized treatment, the heating operation must be very fast. Hence, inherently the method is very well adapted to large-scale operations, such as are required in the mass production industries.

In this paper I shall attempt to cover one specific field of induction heat-treating, that is, the treatment of internal diameters. Here, a zone of relatively shallow depth, commencing at the bore of a hollow body and extending outward radially, is to be heated so rapidly as to be confined to but a portion of the work piece and then, at the required time, quenched.



■ Fig. 3 – Progressive cylinder heat-treat machine



■ Fig. 4 – Heat head in quenching position

■ Hardness

The purpose of this treatment is to harden the inner diameter locally, so that increased wear resistance, greater strength, and other advantages may be realized. The hardness of the hardened zone will be controlled by the chemistry and structure of the cast iron or steel being treated, by the time and rate of heating, by the differential of the temperature of the heated zone and the adjacent material, by the type and drasticity of quench, as well as by the subsequent annealing and stress relieving.

■ Case Thickness

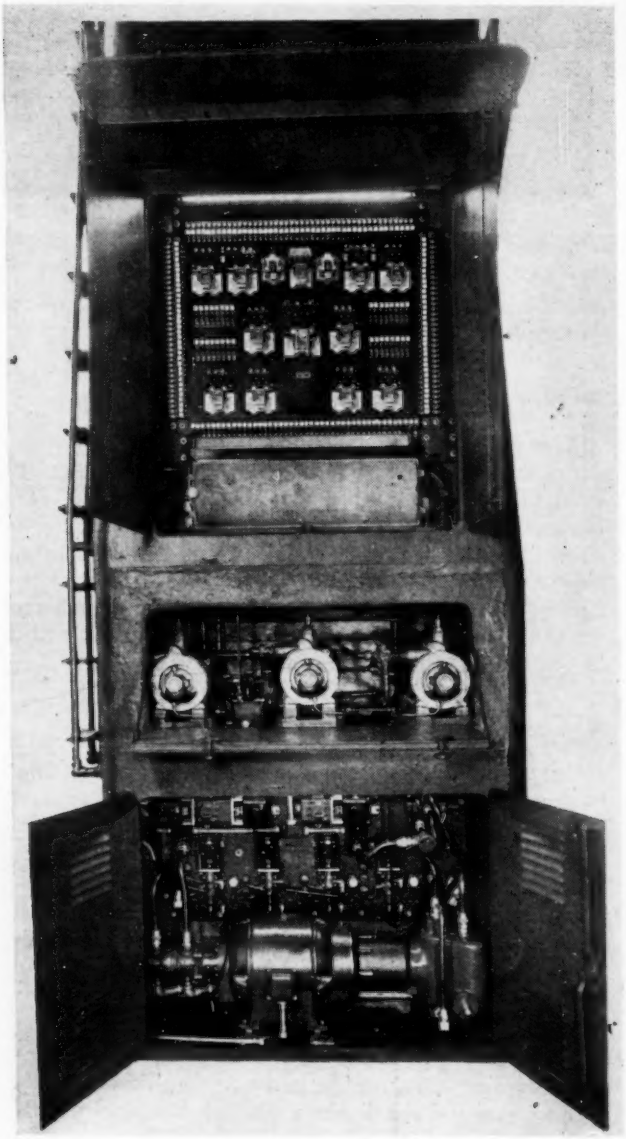
Our definition of case thickness is, that depth of heat-treatment which, prior to any subsequent anneal or stress relief, is predominantly martensitic, being equal in hardness to 52 Rockwell C or more, and in which there has been substantially complete redistribution of carbon.

The desired depth of thickness of this case is determined by the minimum engineering requirements, the susceptibility of the material, and heat-treating method used to produce to the requirements and the practicability of the design.

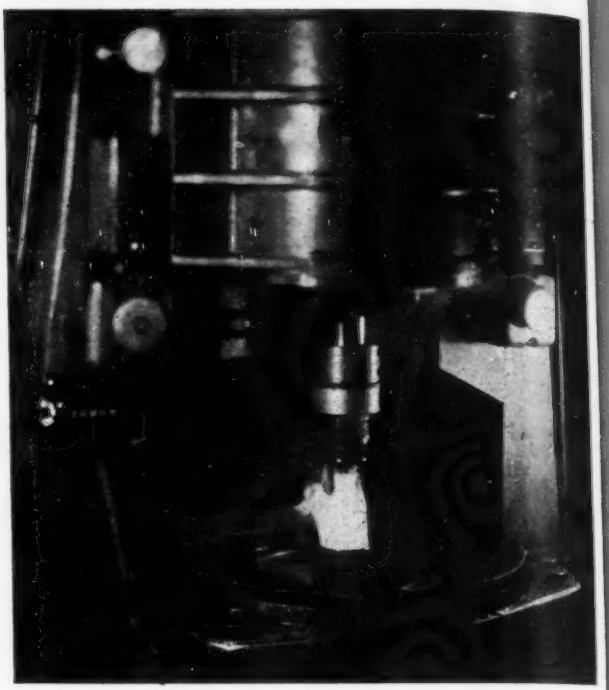
In heat-treatment of the internal diameter of cylindrical bodies consideration must be given to the thickness of the cylinder wall and the depth to which it is desired to heat-treat. In general it may be said of cylinders subject to high internal pressures that not more than 40% of the

cylinder wall should be subjected to high temperature and drastic quench, unless a very thorough job of stress relieving is to be performed subsequently. Where the purpose of the hardening is to provide a close grained, high strength, hard structure, a case of from 0.040 to 0.120 in. thickness may meet all of the engineering requirements and be applied with a minimum of distortion to the cylinder itself.

In order to heat-treat the internal diameter of a thin-walled hollow body successfully, it will be apparent that the heat must be applied at a very high rate for a very short time. It has been experimentally determined that in steel such as SAE 1045 brought up to the required temperature, complete redistribution of the carbon can be secured in a heating time of about 1 sec. Therefore, a very short time may be used for heating, and since the heat is confined to a shallow depth, a very definite differential of temperature will be achieved between the zone to be hardened and the adjacent back body. The back body, having retained a lower temperature and having its strength unimpaired, tends to operate as a chuck holding the part round and straight and also assisting in the subsequent quench.



■ Fig. 5 – Exposed panels showing controls



■ Fig. 6 – Quenching Ford passenger-car hub

It is of great importance that the frequency used should be of such order as to cause induced currents to flow substantially throughout the area in which it is desired to harden, and the absolute minimum of the hardened zone be heated by conduction. This differential heating is accomplished, therefore, through the concentration of high-power, high-frequency currents in the zone to be hardened and to be so concentrated that the temperature of the zone to be heated is raised to the hardening temperature before any substantial amount of heat can drift to the core of the piece.

In the internal treatment of hollow bodies, two methods of applying induction heating are available:

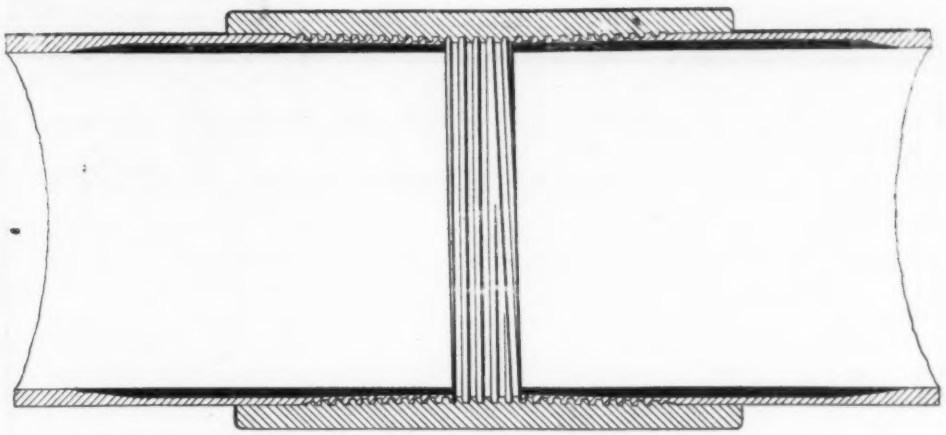
1. To provide an induction heat-treating head designed to apply heat throughout the entire length and diameter of the zone to be heated. The rate at which power must be supplied through induction to achieve the rate of heating required is in the order of 10-25 kw per sq in. of the inside surface to be treated.

2. To provide a narrow heat-treat head and to progress this head through the bore to be heated at a continuous feed rate so as progressively to heat and subsequently to quench the zone to be hardened.

As the power input and the timing of the current application and the volume, pressure, and angle of direction of the quench must be accurately controlled, it follows that a machine must be provided which will accurately control the sequence of the various operations within exceedingly close limits.

■ Stresses

When a cylinder is locally hardened, stresses are produced in the hardened zone and the adjoining unhardened material.



■ Fig. 7 – Reinforced pipe joint

As created, they may be either compression or tension and may be high or low in degree. The value of this trapped stress is influenced by many factors, some of which are:

1. Material.
2. Rate of heating.
3. Depth of affected zone.
4. Rate of quench.
5. Geometry of the piece.
6. Ratio of the heated zone to the wall thickness.

■ *D/T* Ratio

Most commercial hollow bodies have a high *D/T* ratio. This is particularly true of cylinder sleeves where the *D/T* ratio will range from 10 to 1, to 20 (or more) to 1, and dry liners will range from 20 to 1, to 50 to 1 (or more).

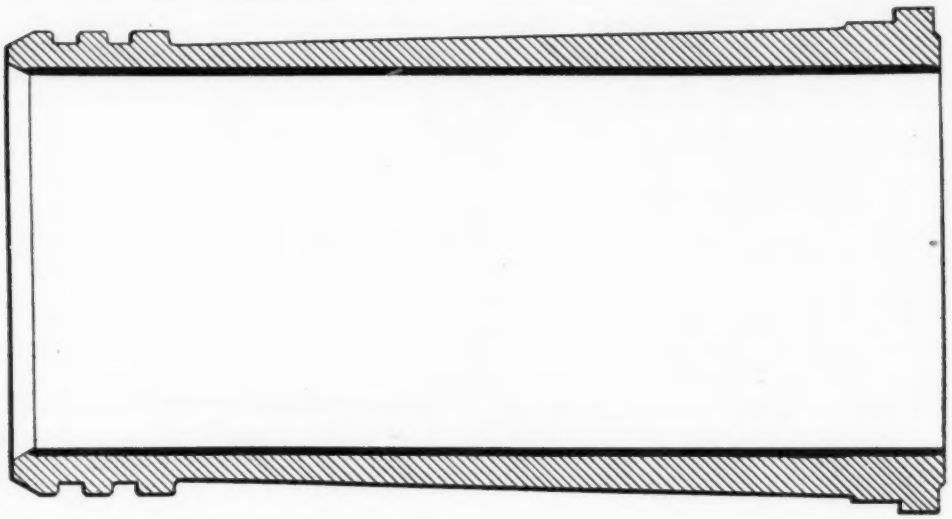
■ Heat-Treat Machine

To apply induction heat-treatment to tubular bodies, it has been found necessary to locate the work accurately so that when the work piece is brought into inductive relation to the heat-treat head it will rotate concentrically about the heat-treat head and quench. For long cylinders, progression of the work piece in relation to the heat-treat head and quench must be provided, and this necessitates two sets of chucks, one on either end of the cylinder. For production equipment, these various operations should be and are performed automatically and mechanically, and are controlled by means of sequence controllers capable of very close adjustment with long life and very accurate performance. During the past 10 years, a number of types of internal induction heat-treating machines have been developed to accommodate various types of work. They may be divided essentially into two general classifications as follows:

1. The static or one-shot classification (Fig. 2): A heat-treat machine wherein the area to be heat-treated is all heat-treated at one time. Such a machine is applicable to heat-treating areas in a bore less than 5 in. in diameter and 6 in. long. It requires large generator capacity in the large bore dimensions but will produce at an extremely rapid rate, heat-treating areas 50 sq. in. or less.

2. The progressive, or scanning, type of heat-treatment

■ Fig. 8 – Liners used in Caterpillar Tractor Co. diesels

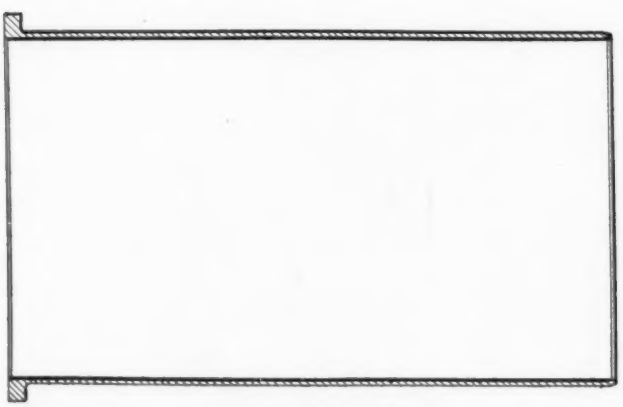


■ Fig. 9 – Cylinder liner

(Fig. 3), which usually is more desirable where the body is cylindrical in form, open at both ends, and of considerable length, say from 6 to 100 in. or more. These machines are equipped with heat-treating heads of relatively short length, that is, from 1½- to 3-in. windows, and normally heat-treat at rates of from 1 to 3 in. per sec. A quench head is arranged to quench the work piece immediately upon the heated zone reaching the proper relation with the quench head. These machines are coordinated assemblies – precise mechanisms.

Too great detail is not permissible at this time, but a brief description of the machine's component parts, source of power, and controls should prove of interest. These are as follows:

1. Source of energy, metering equipment, conductors, and the like: The source of high-power, high-frequency energy consists usually of a motor-driven inductor alternator. The motor, of course, may be of either the induction or synchronous type, depending on the type of power available. It is usually mounted on a common base and directly connected through a flexible coupling to the inductor alternator. The generators installed at our Detroit plant develop alternating current at 9600 c, and have a capacity of 700 kw at 800 v.



■ Fig. 10 – Cylinder sleeve

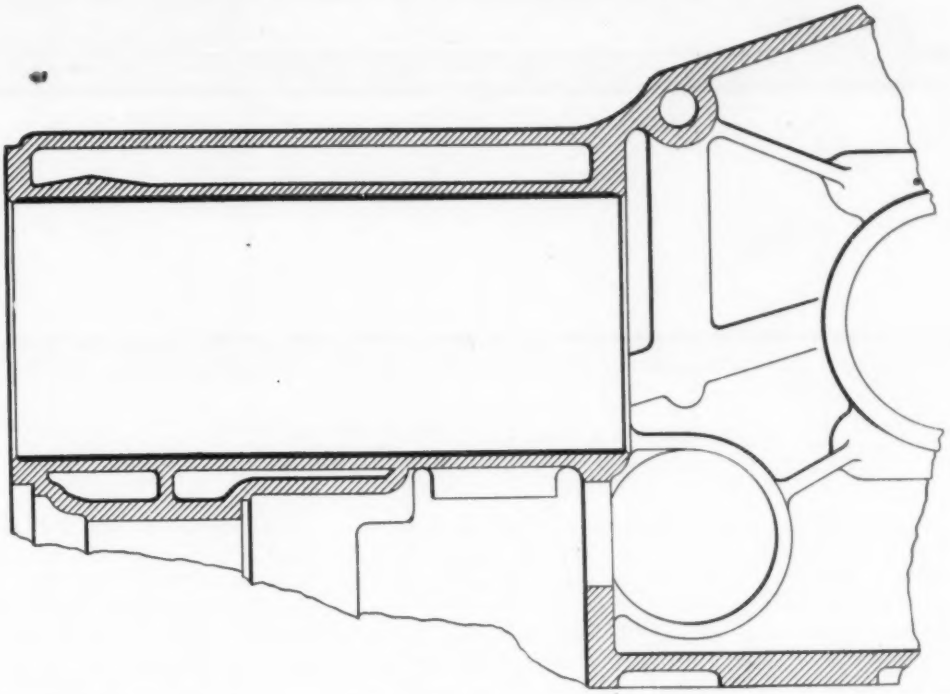
Necessary exciters are furnished and may be driven on the common shaft or may be separately motor-driven. The switching equipment for the high-frequency power consists of a solenoid or air-operated contactor.

Metering equipment comprises the necessary a-c voltmeters, ammeters, power factor indicator, indicating wattmeter and, in some cases, a variometer. The generator may be mounted in a position remote from the induction heating apparatus. The current is conducted from the generator panel to the heat-treat machine, usually by means of a two-conductor, lead-sheathed concentric stranded cable. Such conductors reduce the reactance of the circuit, and with them we transmit this high-frequency power over distances of from 500 to 600 ft with a power loss in the order of only 1-2%.

At the heat-treat machine, the cable terminates in a suitable pot head and the connection is made to a sliding core switching transformer of special design. This transformer serves the dual purposes of (1) reducing the voltage to safe working voltage at the heat-treat device and (2) accurately and safely switching this high power on and off the heat-treat head. This is done through a vertical reciprocation of the transformer primary to remove it from inductive relation with the secondary during the off-load period, thus reducing current flow to the very small exciting current flowing through the primary windings. The load is applied by moving the primary into inductive relation with the secondary, causing current to flow through the work circuit. (See Fig. 4.)

A massive conducting arbor similar to the spindle of a machine tool connects the heat head with the transformer secondary. This conductor arbor is made of two concentric copper tubes electrically insulated from each other by means of a suitable insulating bushing placed between the inner and outer conductor tube. Connection is made between this conductor and a bank of tuning condensers by means of a similar concentric arbor. Transformer and all high-voltage equipment are installed within a compartment. It may be noted that the transformer is placed at the top of the machine, the arbor and heat-head being suspended and mounted precisely from the top face of the frame.

A suitable work-holding fixture is provided and in this



■ Fig. 11 – Automotive cylinder block

particular machine the work piece is simultaneously rotated and progressed axially in relation to the heat head during the heating and quenching cycle. In operation, it is necessary that the product be chucked so as to be held in precise relation with the heat head but not in contact with it. It will be observed that the arbor must be straight and in perfect alignment with the work holding fixture, since the work is rotated around the head and at the same time progressed axially with respect to the head.

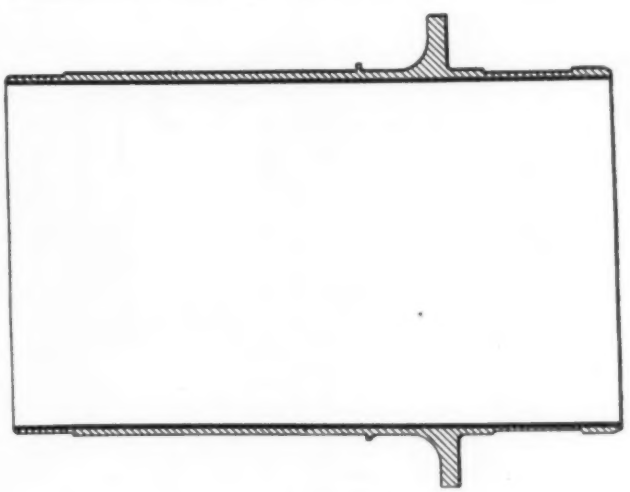
2. The heat-treat head and quench: The heat-treat head, after all of the necessary motivation and sequences have been ensured, is the vital part of the machine. This head consists of a tubular copper coil wound about a laminated iron core, all of which is mounted on a mandrel. This tube, being hollow, permits the circulation of coolant through the coil. Since these tubes are of small cross-section, the orifice is, perforce, restricted and great care is taken that no obstruction to the passage of the coolant develops, since this is operated in some cases up to a current density of 200,000 amp per sq in.

The quenching device is placed directly below the heat head, being maintained in definite relation to it. For purposes of loading this machine, in which the quenching fixture rises from below the head, the quenching device is retracted (that is, lowered) from this relation during the loading and unloading period. This is accomplished by means of an air or hydraulic cylinder operating in timed and controlled relation to the other operations.

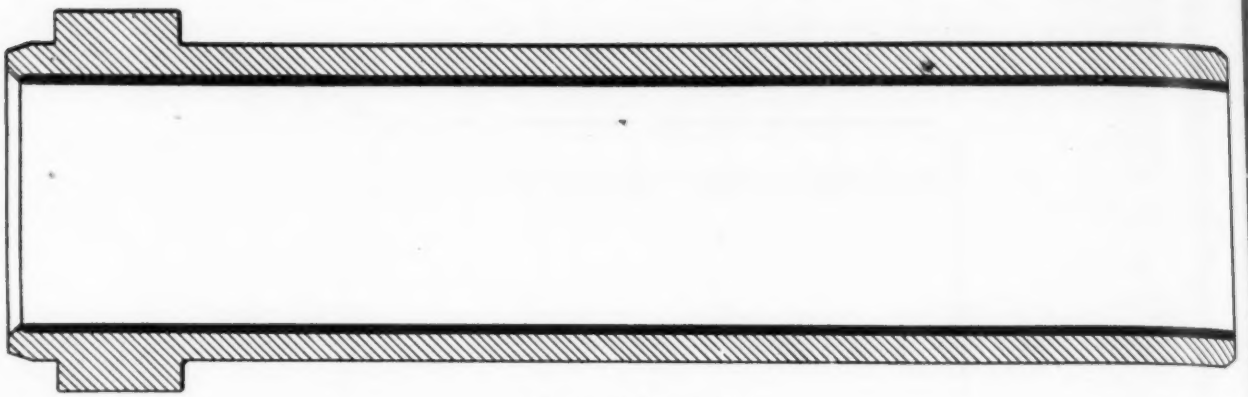
3. The controller: The accuracy of the timing and control of the entire operation is vital.

All of the timed sequences are maintained by means of a sequencing controller consisting of a series of precision cam-operated switches. The drum on which the cams are mounted is arranged in this case to receive four sets of cams, the different sets controlling the operations for different sizes of cylinders to be heat-treated. This controller is designed to give extremely high fidelity in timing of the controlled operations, and also to permit changeovers for treatment of different sizes of bores within a minimum amount of time. (See Fig. 5.)

The controlled movement of the work-holding fixture and the heat-treat head is accomplished by a suitable motor-driven hydraulic system. The transformer primary, secondary, and all connecting arbors are water-cooled. Cooling of the transformer coils is accomplished by passing a metered amount of coolant through both the primary and secondary, the quantity being accurately controlled by means of several calibrated flow control switches. These switches automatically interrupt the control circuits on the transformer so that power cannot be applied to the heat head until the normal flow has been restored.



■ Fig. 12 – Aircraft-engine cylinder



■ Fig. 13 – Slush pump liner

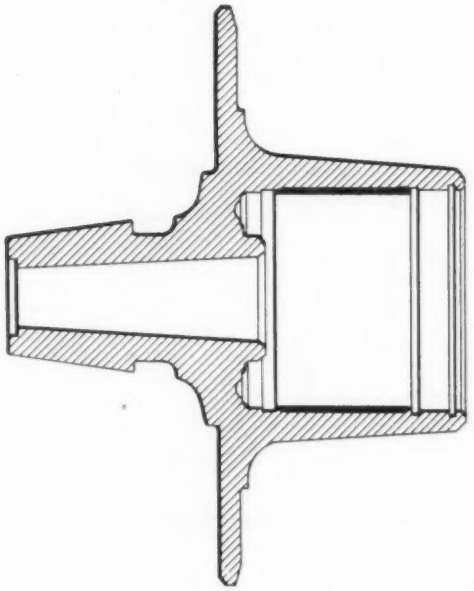
In order to avoid rust or corrosion, a great deal of stainless steel and bronze are used in the construction of the machines. Frames are carefully fabricated of steel or cast iron and thoroughly painted.

Size of the machine is such that not more than 40 to 60 sq ft of floor space are required, which includes the necessary work room around the machine. Such machines are set up and operated in our production line just as any boring mill may be set up and operated and the treated parts are passed on right through to the inspection bench. No accumulation of stock is required for this operation, as we are able to heat-treat the product, inspect, grind, and hone as fast as the line produces.

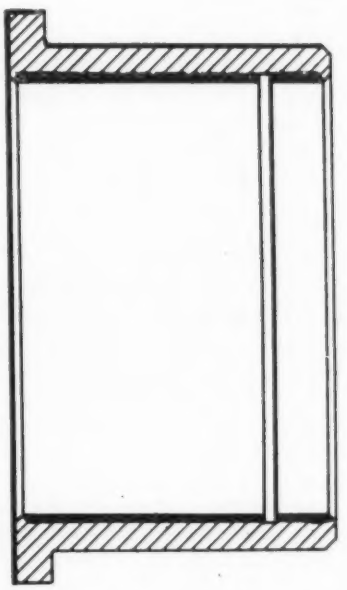
Since the original machines and method, many new methods and machines have been developed. These permit the heat-treatment of engine cylinders of many sizes, and tubes and pipe of longer lengths. Machine tools and fixtures to accommodate objects of many kinds and sizes have been developed.

4. Operation of the machine: The setting of the machine for any given heat-treating operation within its range is, as was noted above, quickly and easily accomplished by adjusting the controller and its cam-operated switches. Once this has been done, the correct heat head and work holding fixtures are in place and adjusted for the particular hollow object to be treated (cylinder, hub, tube, or whatnot), the operation is quite simple.

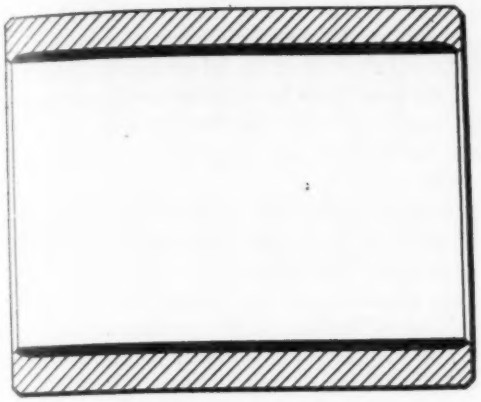
Since the machine is fully automatic, the only requirement is that the operator place the piece to be hardened in the holding fixture. In the progressive type of machine, when this is done, the fixture closes and automatically elevates the work piece (a cylinder, we'll say) to a position in which the heat-treat head and the quenching fixture are brought into relation with the lower end of the cylinder. The cylinder is then progressed downwardly and rotated in relation to head and quench, the power is automatically applied, the heat is produced, with the quench following at the exact moment necessary to pro-



■ Fig. 14 – Wheel hub with straight bearing race



■ Fig. 15 – Driving pinion bearing cage



■ Fig. 16 – Bushing

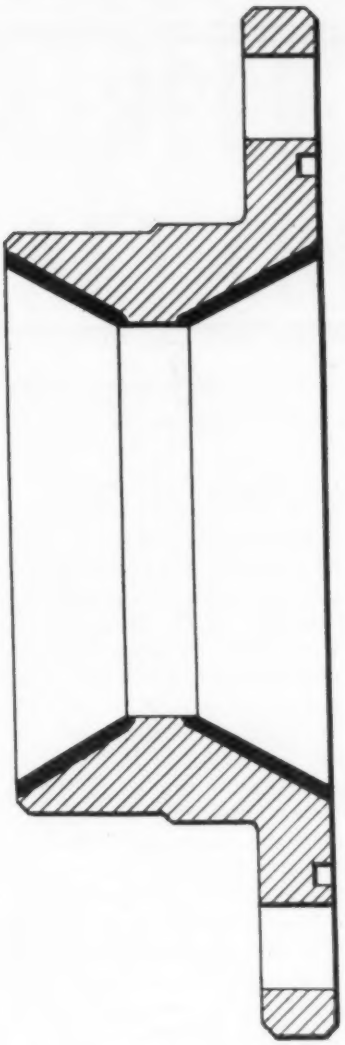
duce the desired hardening. It will be noted that the work is heated progressively and as progressively quenched.

Both of these operations are very precisely controlled, the quench continuing until the necessary heat has been extracted from the part. It is then shut off automatically and moves from its relation with the heat-treat head, when the part under treatment is removed.

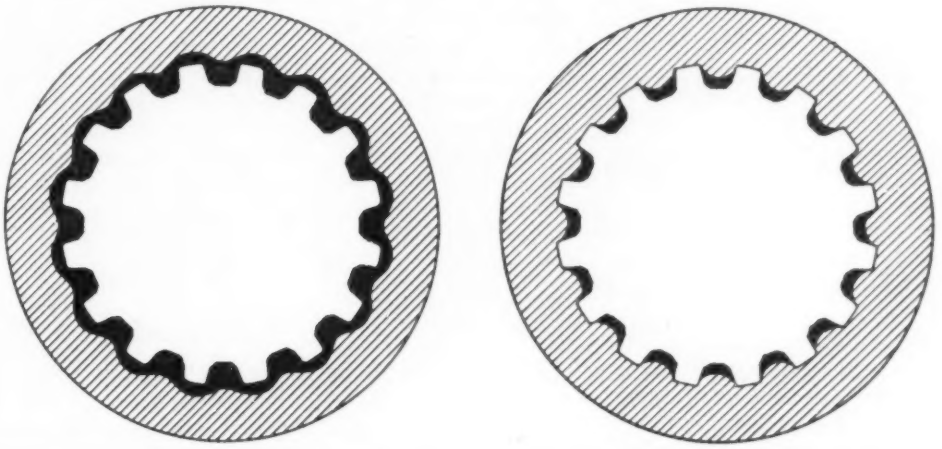
This operation is reproduced periodically as indicated by the sequencing controller and the capacity of the machine.

There have been a number of applications of internal induction heat-treatment to industries during the past eight or nine years. The first application was to the Ford passenger-car rear hub. This part was heat-treated by the "one shot" method. Millions of these hubs were produced with outstanding success, resulting in lowered manufacturing cost and improved quality of product. The material being used for this hub ranged from cast steel to forgings of SAE 1040 and 1045. (See Fig. 6.)

The second major application was to the production of wet cylinder sleeves for a well-known diesel-engine manufacturer. These cylinders were of cupola cast iron, and initially the outstanding advantage was that of reduced manufacturing cost; but it was found that increased performance resulted as well. Over a million of these cylinders have reached ultimate service in all climates and under all conditions found in the world-wide war.



■ Fig. 17 – Flanged bushing



■ Fig. 18 – Internal splines

■ Fig. 19 – Air compressor cylinder

Another interesting and probably vital application was to the ends of oil-well casing, where approximately 10 in. at each end of the casing was heat-treated internally before threading. The threading is performed after heat-treatment and the heat-treat zone is held to very closely spaced relation with the root of the thread without any interference with the subsequent manufacturing operation. An improved threaded end resulted in an increase of collapse strength of approximately 75-85%. (See Fig. 7.)

Many other applications have been made since the commencement of World War II, but it will be necessary to wait until some future date to disclose them.

One of the most interesting developments which grew out of our experience in applying the induction heating process to the heat-treatment of internal diameters was the fact that the process actually, automatically gives 100% inspection of bores treated. This most useful dividend from the process is now regularly being employed in production. What happens is that when a defective casting or forging undergoes the process, porous or cracked sections provide abnormal resistance to the flow of high-frequency current. This in turn produces excessive temperatures which either cause actual melting or other physical signs of overheating, thus quickly showing up defective parts and causing their rejection.

■ New Applications

Because of the facility with which low cost metals, that is, low alloy steels and cast iron, can be utilized by employment of induction heating, the process has been of invaluable aid in the provision of weapons and munitions needed to fight World War II. Naturally, the disclosure of what these are and how induction heating helped will have to await the lifting of Government censorship.

These things, of course, are accomplished facts, and so it may prove more profitable for us to turn to speculation as to how the induction heating process can be employed to help provide better products at less cost for the automotive industry when that great industrial giant swings back into peacetime production and begins to fulfill its promises of better vehicles for the postwar world. It is not exaggeration to predict that great things may be expected through induction heating in this postwar period. I wish to suggest some of the applications of the process which we expect may prove fruitful. Before going on to this discussion, however, I would like to inject a few suggestions which may help all of us to take fullest advantage of a production tool which we consider one of the most useful developed in recent years. We venture to offer them because of the great interest the automotive industry has evidenced thus far in the advantages of induction heating. If we bear them in mind as we approach our problems I am sure all of us will benefit.

Stated briefly here are some of the things experience has taught us:

1. Successful application of induction heating requires a background of experience plus a wide knowledge of the process and its capabilities.

2. Full advantage of the process can be gained only after exhaustive analysis of the product sought to be improved. If necessary, the product should be redesigned, with design engineers and induction heating engineers collaborating, in order to improve quality and reduce cost.

3. Through the use of induction heating a combination of parts may be achieved which will be an improvement over previous designs.

4. The process offers valuable opportunities for wider use of low-cost materials.

And now, with these suggestions out of the way we can pass on to a discussion of some of the applications of induction heating of internal surfaces to problems of the automotive industry. Those we will discuss are those which experience has shown may be of considerable importance. Many more – some of them, no doubt of even greater importance – must wait until the production and design engineers discover them.

■ Cylinder Sleeves

Where wet cylinder sleeves can be used, their production and induction heat-treatment are relatively simple. These cylinders may be made of cast iron or any other suitable material, such as malleable iron, cast steel, or steel tubing. All these can be successfully heat-treated at low cost. (See Fig. 8.)

■ Dry Cylinder Liners

Precise cylinder liners of high strength can be internally inductively heat-treated. Careful manufacturing techniques, when followed, will produce accurate liners at reasonable cost. (See Figs. 9-13.)

■ Hubs

Hubs embodying integral bearing races have been manufactured in production, the plain ones having run into millions of pieces. Improved quality and lowered costs have been experienced. One advantage of the integral bearing race stems from the elimination of creep between the bearing race and the hub, with consequent fretting or galling of the hub recess. Many hub bearing race combinations fail as a result of relative movement between the bearing race and the hub. (See Fig. 14.)

■ Integral Bearing Cages

A good example of such a cage is the one illustrated in Fig. 15, which is used to mount the final drive shaft in the truck differential housing. Here it is obvious that

concluded on page 63

A PETROLEUM ENGINEER LOOKS at AIRCRAFT FUELS

by A. G. CATTANEO, F. G. BOLLO, and A. L. STANLY
Shell Development Co.

IF the refiner is to succeed in blending good, low-cost aircraft fuels from the hundreds of components now available to him, these authors point out that it will be necessary for the aircraft and oil industries to cooperate in determining actual fuel performance requirements.

To help reach this goal, the authors present here an extensive discussion of the factors affecting the performance of the fuel-engine combination; namely, power, economy, and reliability (including safety).

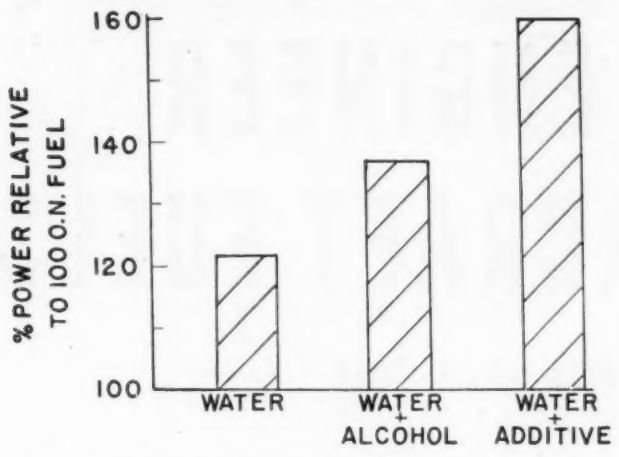
THE AUTHORS: A. G. CATTANEO (M '38) joined the staff of Shell Development Co. at Emeryville, Calif., in 1937, and organized a motor laboratory which is now carrying on investigations on the development and application of engine fuels and lubricants under his direction. Dr. Cattaneo, who received his engineering education in Zurich, Switzerland, and Karlsruhe, Germany, served as a member of the Delft Research Laboratory of the Royal Dutch Shell Group from 1934 to 1937, where he did research on combustion phenomena in gasoline and compression-ignition engines. He is a past-chairman of the SAE Northern California Section. F. G. BOLLO (J. '43) has been research engineer at Shell Development Co. since his graduation from the University of California in 1938. He is now engaged in research on aircraft fuels at the motor laboratory. A. L. STANLY (J. '42) has been concerned with research on various performance characteristics of fuels and lubricants used in reciprocating and turbine engines at the motor laboratory of Shell Development Co. He has been with the organization since graduating from the University of California in 1938. Mr. Stanly was 1944-1945 vice-chairman for aeronautics for the SAE Northern California Section.

DURING the war great strides have been made in the performance of aircraft engines, and the newer developments promise strides of equal importance during the next decade for military as well as commercial application. In this progress the job of the oil industry can be defined as that of making better fuels. It sounds simple, and seems to require research only with respect to fundamental chemistry and applied engineering of refining processes. But there is another question which must be answered before the chemist can even start on his work. That question is: "When is one fuel better than another?" Just a few years ago that question could be answered by simply comparing the octane numbers. But today for conventional engines we demand other properties also, particularly rich mixture performance, which are not indicated by octane number at all. In fact most of our 100-octane-number fuels of the 1930's would be entirely inadequate in present-day engines, which operate very satisfactorily on modern 100-octane-number fuel. So the octane scale is only a partial answer in the conventional engine, and for the turbine powered engine octane number does not constitute a quality scale at all.

Yet the refiner must understand performance requirements. His problem is not as simple as selecting one fuel from half a dozen. Currently, he produces some 20 different materials as potential components of aircraft-engine fuel and he could produce literally hundreds of others. Thus, he has to decide which of these to blend, and in what proportion. It is true that today the Government to a large extent tells him, in the interest of the greatest overall national supply, how to do it. But very soon he will have to make his own choice from the thousands of possible mixtures.

If only the success of each particular oil company were at stake, the aircraft industry could afford to disregard the problem. Actually, there is a national stake, whose disregard in prewar days brought us close to danger in 1942. Up to that time only paraffinic fuels had been used in this country, and this preference was written, directly or indirectly, into all specifications used by the engine manufacturers, airlines, and the Armed Forces. The reasons for this preference were largely traditional. Engine tests had shown us and others engaged in this field of petroleum research that aromatics could impart to the final blend the rich mixture performance quality which to this day cannot be equalled by any commercial blend without aromatics – and that with only minor variations in some relatively quite unimportant fuel properties. However, in our competitive economy a customer can stand by his preferences, whether they are justified or not, and as the quantities of fuel were minute – only some 250,000 gal per day for the whole U. S. – they were easily furnished by the oil industry. Then came Pearl Harbor and the

[This paper, with the title "A Petroleum Research Engineer's Outlook on Fuels for Conventionally Powered and Gas Turbine Powered Aircraft," was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Aug. 24, 1945.]



■ Fig. 1 – Detonation-limited power with internal coolants on modern aircraft engine

sudden demand for a rapidly rising supply – over 25,000,000 gal per day in 1944. And it turned out that only by the use of the special properties of aromatics could this quantity of fuel of the requisite rich mixture performance be obtained. Without aromatics we would even today have only perhaps 10,000,000 gal per day of fuel of current quality. So it was necessary suddenly to switch our aircraft operations over to fuels containing aromatics. While this was successfully accomplished and gave major trouble only to the rubber people, who had to find new materials to replace every rubber part in the fuel system, one shudders to think of the consequences that such disregard of the national supply position might have some other time.

There are a great many people who feel that such a situation can be avoided only by constant contact and technical cooperation between the aircraft and oil industries. It is in the interest of both industries to make this cooperation voluntary. There would be gains apart from the above discussed national interest. For example, the aircraft industry could be assured that fuels of the quality for which their airplanes have been developed can be obtained at any point on the globe with the most economical production and transportation. In the coming days of transoceanic traffic, cheapness and general availability of good fuel should materially help the progress of aviation.

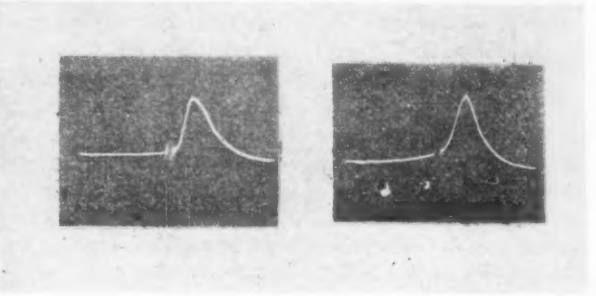
It is the purpose of this paper to analyze fuel quality in terms of performance of the fuel-engine combination. There are only three performance qualities that are demanded from any fuel for any type of engine:

1. Power.
2. Economy.
3. Reliability (including safety).

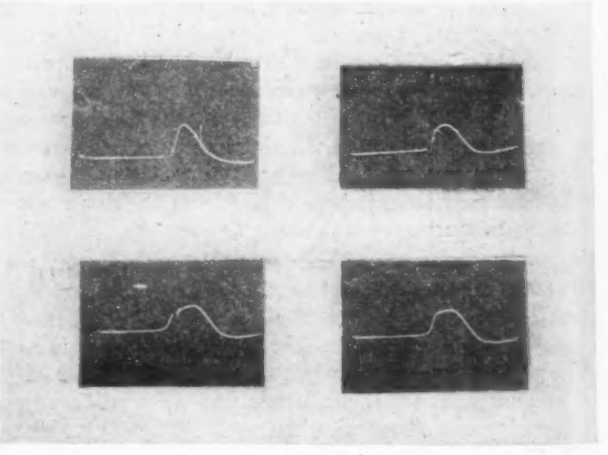
■ Power and Economy – Reciprocating Engine

In the reciprocating engine, power is usually limited by one of two factors: the cooling capacity of the engine, or detonation. If, in any particular type of operation, the waste heat generated in the combustion chamber is larger than that which can be absorbed by the cooling medium, temperatures will rise, and failure will result because of the reduced mechanical strength of engine parts. If, during this temperature rise, preignition sets in, that is, untimely ignition of the charge by overheated parts, the inefficiency of the combustion will accelerate the temperature rise further, usually with disastrous results. However, in the early stages the temperature rise is quite slow and acceptable if kept safely below preignition conditions. It is in this range that take-offs are made, safeguarded by a maximum for cylinder-head temperature and by a time limit. The cooling ability limits particularly the climb, which is of long duration and therefore requires equilibrium conditions, that is, constant temperature, even though cooling is limited by the low air speed and aerodynamic requirements (closed shutters). Thus, the majority of present-day installations of aircooled engines are cooling-limited under high-power conditions when operated on 100/130 grade fuels, and even in some cases on 91/96 grade fuels. In liquid-cooled engines, detonation is more likely to be the critical factor.

To overcome this cooling limitation, water injection is used in addition to the extra cooling obtained by enrichening the mixture. Water also suppresses detonation, and where the cooling and detonation limits are reached at about the same time, its use will lift both limits. This fact is illustrated in Fig. 1 for a current engine. Where detonation is more limiting than cooling, it is possible to add detonation suppressing agents to the water, and in this way brake mean effective pressures that exceed the



■ Fig. 2 – Supercharged CFR engine combustion diagram – left: normal combustion; right: detonating combustion



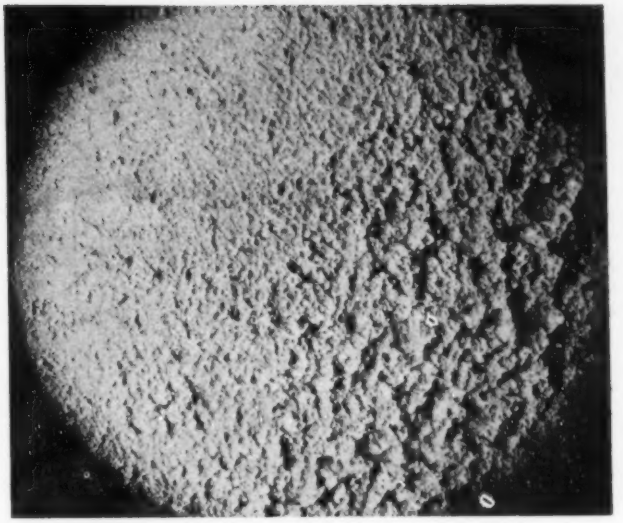
■ Fig. 3 – Supercharged CFR engine combustion diagram – upper left: normal combustion; upper right: mild preignition; lower left: advanced preignition; lower right: advanced preignition – no spark

mechanical strength of any engine can be reached without detonation.

In take-off or climb, detonation is probably not as much a cause of power loss or engine failure as often thought. In a representative, modern, large, aircooled engine a manifold pressure 5 in. Hg higher than that giving incipient detonation gave only a minor increase in the intensity of detonation. It would have taken a still higher manifold pressure to obtain detonation so severe that power loss or rapid piston failure would result. It is unlikely that a pilot's error, or the severity of a particular take-off, should exceed so wide a range if fuel and engine are anywhere near normal. Far more often power loss in take-off is attributable to an excessively rich mixture, misfire, or other factors. Similarly, actual engine failure in take-off is more often due to overheating or preignition induced by a defective spark plug or other part, than to detonation.

In detonation the only effect of importance to engine operation seems to be the mechanical one of the steepness of the pressure rise, and of the subsequent oscillations of the combustion-chamber gases. Instead of gradually applied pressure we have the mechanical effect of hammer blows and bouncing of the hammer, as shown by the combustion diagram in Fig. 2, and the accompanying surface temperature oscillations. Thus, the fatigue limit of the walls surrounding the gas is reached correspondingly faster. That preignition is not necessarily accompanied by detonation is shown by the absence of the characteristic detonation vibrations in the preignition cycles of Fig. 3.

The effect of these hammer blows will be first noticed on certain parts of the piston surface, which have the lowest strength because they consist of aluminum and are at the highest temperature. A typical failure of this type is shown in Fig. 4, and in greater magnification in Fig. 5. In a preignition failure (Fig. 6) the characteristic surface erosion is absent. In addition, under severe detonation, the gas vibrations sometimes excite ring flutter, with its consequences of rapid wear and overheating due to a lack of sealing and blowby.



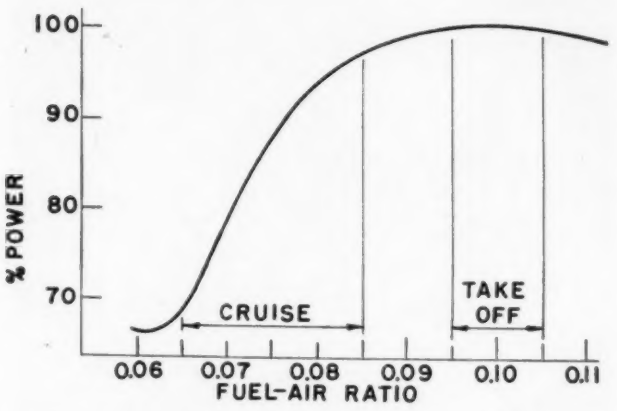
■ Fig. 5 – Typical detonation injury – enlarged view

If the piston-surface type failure is interpreted as a cumulative effect, it becomes obvious that the moment of failure may be hundreds of hours removed from the particularly severe operation which contributed the largest share, or that the failure may be due to an accumulation of many hours of cruising in the detonation range. Thus, prevention of such failures is as much an engine maintenance as a flight engineer's responsibility.

Economy in the conventional engine means the use of the fewest pounds per hour for a given horsepower, or in other words the fuel with the highest Btu content on a weight basis. If we attempt to use such fuels at the most economic, that is, lean, mixture ratio, we find that detonation is the limiting fuel factor. Very little reliable informa-

■ Fig. 4 – Typical detonation injury on supercharged CFR engine aluminum piston

■ Fig. 6 – Preignition failure on supercharged CFR engine aluminum piston



■ Fig. 7 – Typical detonation-limited mixture response curve

tion is available today on the practical significance of this factor. We have operated a large engine through hundreds of hours of fuel tests at a moderate detonation level without apparent ill effects to the engine. It is only recently that really good instrumentation for the measurement of detonation in flight has become available, and it may be hoped that more exact information will be obtained soon. However, such information may not really help very much because of the peculiarities of fuel distribution. Our at present most common system of injecting fuel into a central manifold results in inequalities of the individual fuel charges that reach each cylinder with regard to quantity of fuel and composition of fuel, particularly tetraethyl lead content. These inequalities will further change with every operating variable, particularly with engine speed and air temperature. In rich mixture operation such inequalities affect the detonation limit relatively little. In lean operation, however, they change the detonation limit very substantially, as is evident from the shape of the typical mixture response curve (Fig. 7). Thus, in lean operation we must, in fuel quality as well as in operating conditions, cater to the most unfavorable cylinder. Fuel injection of the unmodified fuel in the appropriate quantity into each individual cylinder or cylinder manifold would overcome this difficulty, but it is realized that this advantage must be weighed against a loss of supercharger volumetric efficiency.

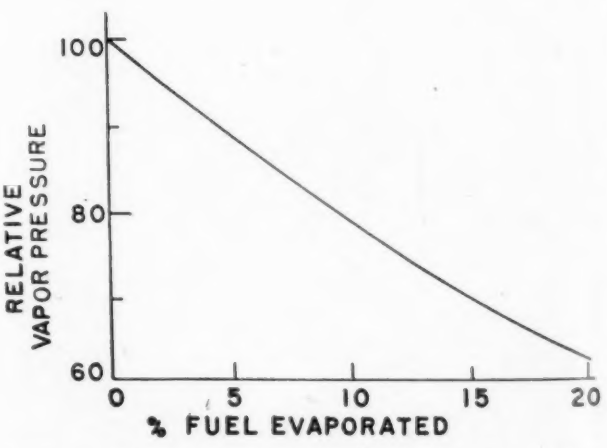
The above concept of economy as pounds fuel per horsepower-hour applies only to a given airplane and flight condition. A more fundamental concept is that of pounds fuel per mile, which introduces the relationship between thrust horsepower, speed, and altitude, and may well lead to the use of high-power cruising even for long ranges, particularly where high speeds command a premium. In this case, as in others which increase the detonation requirements of the fuel, three ways can be followed: either a fuel of higher lean mixture antiknock value, and therefore of higher cost, can be used to permit cruising at the best economy mixture; or a lower grade of fuel can be used at a richer fuel-air mixture; or the lower grade of fuel can be used at the best economy mixture in conjunction with water injection with or without additives. The relative economy of each of these three methods will have to be evaluated for each case, with due consideration being given to cooling limitations of the engine installation, relative fuel cost and its importance, and so on.

■ Power and Economy – Gas Turbine Engine

In the gas turbine engine detonation, which is characteristic of batchwise combustion cycles, has no counterpart because of the continuous combustion, and the concept of octane number in its various forms has no significance. Power in this type of engine is limited by the maximum permissible turbine blade temperature. It is known that fuels of widely different character can have very different combustion temperatures, because of differences in dissociation phenomena and in specific heats of the combustion products. In the reciprocating engine these differences affect peak temperatures and pressures. In the turbine engine at the lower temperatures permitted at the blades, dissociation itself, and therefore differences in dissociation, are negligibly small. Thus, these differences in fuel composition will have practically no effect. However, if blade temperatures should ever reach the neighborhood of 2000 F or above, the effect may become noticeable.

On the other hand, fuel composition does affect the flame speed, and if the fuel is not completely burned before it has passed the turbine or even the tail pipe, the efficiency of the engine will be lowered. In addition, intermediate combustion products are always likely to be chilled on cold walls and form deposits which restrict heat flow through the wall, much as they do in conventional engines. In regard to deposit formation a fuel property of importance is likely to be the carbon-hydrogen ratio. It has long been known to be important for deposit formation in various types of combustion systems ranging from the kerosene lamp to the high-speed gasoline engine. The degree of importance of this fuel characteristic for any new type of combustion chamber, such as that used in turbines, will depend upon the geometrical arrangement, temperatures of the walls, speed and turbulence of the air stream, rate of mixing and vaporization of the fuel, and possibly other factors. Thus, engine design will largely determine the permissible limits.

The volatility of a fuel will also have considerable in-



■ Fig. 8 – Change in vapor pressure of aviation gasoline resulting from evaporation

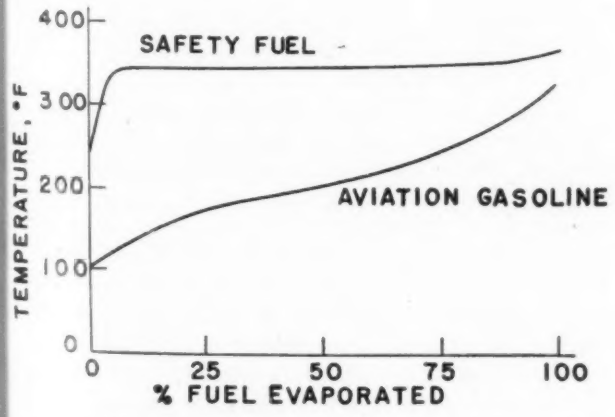
fluence on the formation of combustion-chamber deposits, since nonvaporized fuel tends to polymerize or crack to solid materials. The gas turbine can use heavier, that is, less volatile, fuels than the conventional carburetor engine because the fuel can be atomized under pressure and is sprayed into an air stream already heated by compression. How heavy the fuel can be will again depend upon combustion-chamber design.

In gas turbine engines also, economy means using the fuel with the highest possible Btu content. But, whereas in propeller-driven airplanes weight is always at a premium and space for fuel is amply available in the wings, the very thin wing profiles for high-speed jet-propelled planes offer no space for fuel, and the premium may shift from weight to space. In that case the fuel should have a high Btu content per gallon rather than per pound. As Table I shows this would make a very different type of fuel component desirable.

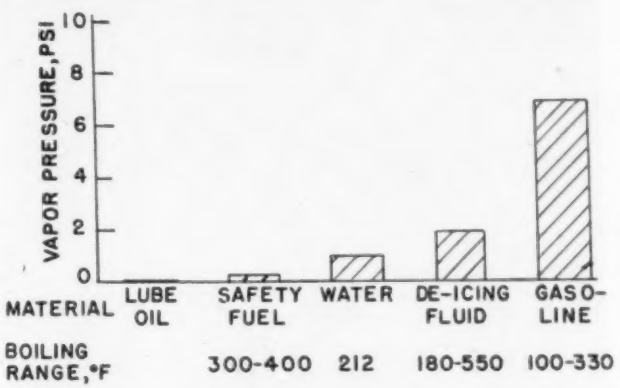
■ Reliability

We require of aircraft fuels, in addition to power and economy, a high degree of reliability. By this is meant here freedom from any undesirable effect on engine operation and the least possible hazard. These requirements cover such things as vapor lock and evaporation losses in flight, ease of starting, good manifold distribution, and the fire hazard. Consideration of these items will show that fuel volatility, that is, the properties that determine fuel vaporization, is one of the major factors involved in each. Before practical aspects are considered, a brief discussion of the concepts of volatility which are involved might be helpful.

Fuel is a mixture of many hydrocarbons which have different individual vapor pressures and boiling points. When it is heated the volatile hydrocarbons boil away first and the vapor pressure of the remaining material drops. This is shown in Fig. 8 for a conventional gasoline. In Fig. 9 the change in boiling temperature, which accompanies the drop in vapor pressure, is shown by the regular ASTM distillation for the same gasoline, and for a fuel boiling in the 300-400 F range – a type of fuel often designated as "safety fuel." A similar change in vapor pressure



■ Fig. 9 – ASTM distillation of safety fuel and aviation gasoline



■ Fig. 10 – Vapor pressures of some materials used in aircraft

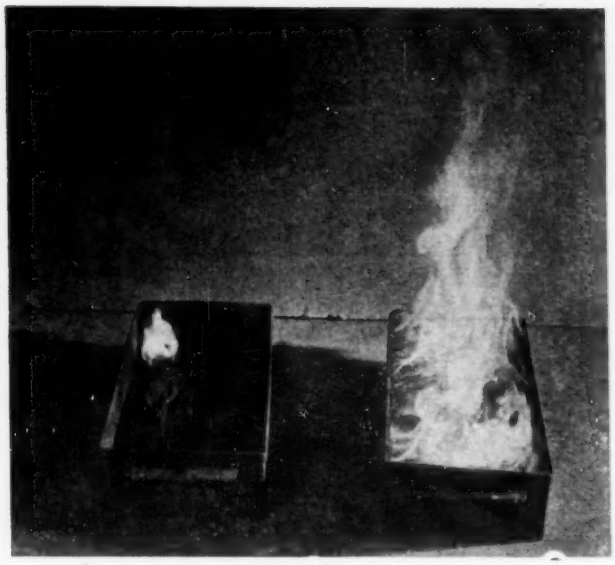
and boiling temperature is obtained if evaporation is caused not by heating the fuel, but by lowering the pressure of the air, that is, by taking the fuel to increasing altitudes.

Table 1 – Heat Content of Various Compounds

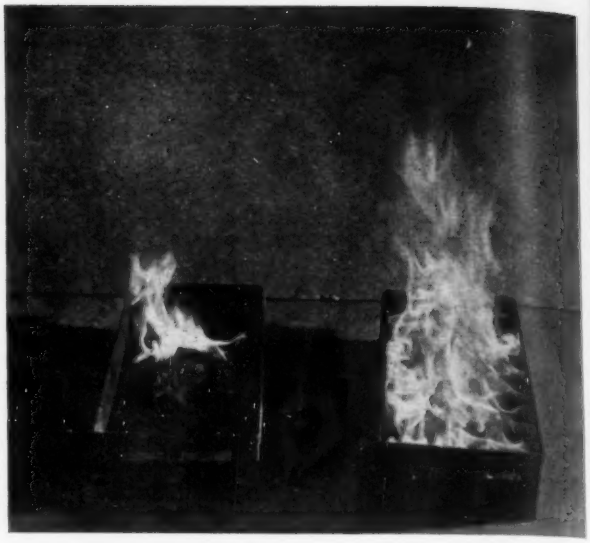
Fuel	Specific Gravity	Boiling Point, F	Freezing Point, F	Lower Heating Value, Btu per lb	Lower Heating Value, Btu per gal	Gravimetric Heating Value (Btu per lb) Compared to Kerosene, %	Volumetric Heating Value (Btu per gal) Compared to Kerosene %
Kerosene	0.81	340-510	−40	18,900	127,100	100	100
Carbon (crystalline)	3.52	...	...	14,520	425,000	77	334
Hydrogen (liquid)	0.07	−256	...	51,608	30,000	529	24
Paraffins							
Pentane	0.628	97	−202	19,700	103,000	104	81
Heptane	0.683	208	−132	19,300	110,000	102	87
Cetane	0.773	543	65	18,900	122,000	100	96
Olefins							
Hexene	0.677	147	−144	19,100	107,000	101	84
Aromatics and Naphthenes							
Benzene	0.878	176	41	17,300	126,000	93	100
Cumene	0.864	309	−143	17,800	127,000	94	100
Decalin	0.888	379	−193	18,300	135,000	97	106
Tetralin	0.974	405	−22	17,500	140,000	93	110
Methyl-naphthalene	1.025	470	−13	17,500	149,000	93	117
Alcohols							
Methanol	0.791	149	−141	9,000	59,100	48	47
Ethanol	0.793	172	−177	11,700	77,000	62	61

The shape of the curves will depend on the composition of the fuel. If a small amount of very volatile material is mixed with a heavy base material, a much more rapid change in vapor pressure with boiling will occur than with a blend of materials more nearly equal in vapor pressure. The most volatile constituent included in the current type of aviation gasoline is butane with a Reid vapor pressure of 51 psi at 100 F and a boiling temperature of 31 F at sea level. At the other extreme are materials like cumene with Reid vapor pressures around 0.2 and boiling temperatures of about 300 F. Most materials included in current gasolines have vapor pressures well below the figure usual for the finished fuel. In order to meet the final value, isopentane is added to raise the vapor pressure. As a rough average the vapor pressure is increased 3 to 4 psi in this way.

We can then, by selecting the materials, control the vaporization characteristics of the fuel over a wide range. For ease of starting and more uniform fuel distribution with current carburetor engines a highly volatile fuel is



■ Fig. 11 – Left: safety fuel 120 sec after lighted match was thrown on surface; right: aviation gasoline 1 sec after lighted match was thrown on surface



■ Fig. 12 – 60 sec after Fig. 11

desirable. On the other hand, fuel of low volatility is required to reduce vapor lock, evaporation losses, and the fire hazard. Some of these factors will be discussed below.

■ Fire Hazard

No petroleum product, and for that matter no material in the world, is inherently safe or unsafe. Rather it is endowed with certain physical properties, and it depends on the circumstances of the use of the material whether these physical properties will contribute to safety or add a hazard. Thus, it is a very real fact that one specific property can contribute to safety under one set of conditions, and increase the hazard under another. If the attempt is stopped to assign a safety rating to a fuel which is applicable to all phases of its use, and instead each aspect of the fuel's use is evaluated in terms of its properties, much confusion in the matter of safety can be avoided.

A. Burning – Petroleum fuels and lubricants in the liquid form combine with oxygen only at an extremely small rate. When they burn, oxygen combines with their vapors. If sufficient air is available, the rate of burning, or the intensity of heat liberation, will depend on the amount of hydrocarbon vapor which is available and continues to be available by supply from the liquid phase. If we look for an index to describe this rate of vapor formation, we find no easily accessible precise one. However, we can use the concept of vapor pressure as a satisfactory guide.

Fig. 10 shows the vapor pressure of some liquids commonly used in aircraft. The vapor pressure of lubricating oil is practically zero, and it is impossible to keep lubricating oil of room temperature burning. Under similar conditions kerosene and the so-called safety fuel will hardly sustain a flame. A 7-psi vapor pressure gasoline, however, has quite enough vapor above it to sustain an intense combustion.

Thus, at ordinary atmospheric temperatures the entire surface of gasoline that has been spilled will almost instantly ignite if a source of ignition is present, and burn with great violence. With a safety fuel, the fuel surface must be heated until the vapor pressure is high enough to furnish sufficient vapor to maintain combustion. Instead of burning instantaneously over the whole surface the flame cannot progress until the adjacent fuel has been heated. For example, if we ignite one spot on a tray filled with volatile gasoline the flame will flash across the whole surface almost instantly. With safety fuel, however, the flame travels very much slower. This experiment is shown in Figs. 11 and 12. Under favorable conditions the safety fuel flame may even die out. If it does progress, however, it may eventually also develop into an intense fire, but this will not happen as rapidly.

In the case of these low vapor pressure materials, much depends on the presence of cooling or heating surfaces. On a hot exhaust pipe lubricating oil may continue to burn, whereas on a cold metal surface the flame would die out. If safety fuel is spilled and ignited, the type of ground and wind conditions will determine the rate at which the flame develops into an intense fire or flickers out. Some soils act as a wick, that is, they permit ready heating of a small quantity of the fuel, and in this way even quite heavy fuels will sustain combustion.

There are still other effects, such as that of splashing, which modify the picture, but in every case the lower vapor pressure leads to a slower progress of the fire, and a lesser intensity of heat liberation.

B. Igniting – The ignitibility of a compound is normally considered to be a property dependent only on the tendency of the vapor to associate with oxygen. However, in practice other physical properties play an important part, mainly the volatility of the liquid, which determines whether enough vapor is present for ignition. In fact, for the normal range of fuel composition, it may be assumed that the tendency of the vapor to ignite is quite independent of its composition. If gasoline is cooled to —70 F, where its vapor pressure is approximately the same as that of lubricating oil at ordinary temperatures, it is just as difficult to ignite. But volatility has another effect beyond that of determining whether enough vapor is present. Consider a liquid splashed onto a hot exhaust pipe of

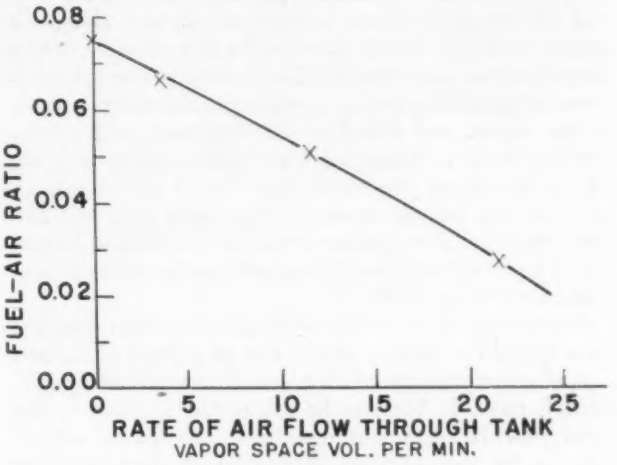
approximately 1000 F. Lubricating oil will wet the pipe, become hotter and finally burst into flame. If the liquid were gasoline, on the other hand, it would start to boil violently wherever it touched the exhaust pipe surface, and its temperature would never rise beyond that of its boiling point, or range. It behaves like water which cannot be heated above 212 F. Thus, the exhaust pipe will not ignite the gasoline. One may well ask why does not the lubricating oil boil and thus keep its temperature constant at a level below that of the exhaust pipe.

The answer is that petroleum products that are too heavy to boil before they are heated to 600-800 F will decompose at these temperatures. In so doing they probably form certain volatile materials that are chemically very active; these immediately react with oxygen, that is, they ignite. Hydraulic fluids behave much like lubricating oil in this regard. Kerosene is borderline. Safety fuel behaves in a manner similar to gasoline.

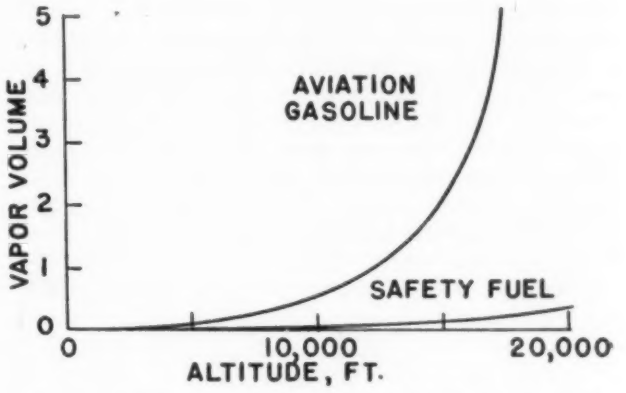
C. Exploding – An explosion can occur when the ratio of air and fuel vapor falls within certain limits. The amount of fuel vapor depends on the vapor pressure of the fuel, and therefore also on the temperature of the liquid. The amount of air present depends on the atmospheric pressure, and therefore on altitude. With a 7-psi gasoline at sea level and at normal temperatures, so much vapor is present in the atmosphere above the fuel that the vapor-air mixture is well outside the explosive range. At higher altitudes this excess of fuel is even greater and the distance from the explosive limits correspondingly larger. For a safety fuel, at sea level and low atmospheric temperatures the vapor-air mixture lies beyond the lean limit of the explosive range, which means that there is not enough vapor in the air to permit an explosion. But as the vapor-air ratio is increased either by decreasing the amount of air, that is, increasing the altitude, or by increasing the temperature of the fuel and so producing more vapor, the mixture falls within the explosive range. Thus, over a wide range of practical conditions, the so-called safety fuel leads to explosive conditions.

The phenomena of explosion and burning should not be confused. Burning, which can take place over a range of vapor-air mixtures much wider than the explosive range, causes damage only by the liberation of heat and its ill effects. Explosion implies the presence of a detonation wave, which cannot disperse itself through vents and other openings, and causes extreme mechanical damage. There are ways in which the formation of an explosive mixture can be prevented. Cooling the fuel before loading will not eliminate this hazard if flight at very high altitudes is considered. One could blow a stream of fresh air through the vapor space of the fuel tank, in order to remove the vapor that is formed before an explosive concentration is reached. The tests shown in Fig. 13 indicate that it might become necessary to change the contents of the vapor space many times a minute to reduce the vapor concentration sufficiently. If this is not practical, there remains the possibility of blanketing the fuel in the tank with an inert gas which contains no oxygen. CO_2 would be the most likely choice. Exhaust gas from reciprocating engines could be used, but exhaust gas from turbines would be less satisfactory because of its high oxygen content.

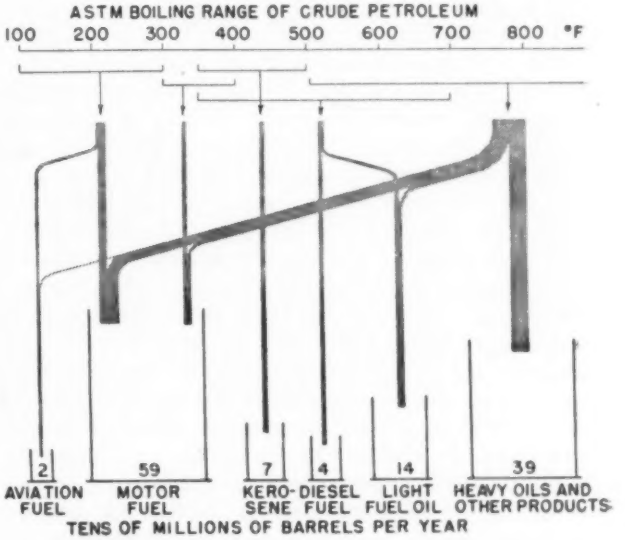
D. Practical Aspects – In the light of the above discussion, it is possible to analyze some of the more common fuel hazards in airline operation, such as the loading of



■ Fig. 13 – Fuel-air ratio in vapor space of aircraft fuel tank with various rates of air dilution – safety fuel



■ Fig. 14 – Relative vapor volumes from safety fuel and aviation gasoline



■ Fig. 15 – Principal use of petroleum fractions in 1940

gasoline for example. If a 7-psi vapor pressure material is loaded the large volume of vapor will flow out the vent and fill openings down to the ground and along the ground with the wind. Even at 20 ft and more, under favorable conditions, the match of a passenger or the spark from the ignition system of a passing vehicle may set flame to this vapor, and this flame flashing back will set the opening on fire. Where lubricating oil or safety fuels are loaded this is not possible as there would not be enough vapor for the purpose unless the fuel were extremely hot. However, with these heavier materials the conditions may just be right for explosion in the tank, and a spark of static electricity may set it off.

Another hazard, very real to our airlines, is that resulting from spilled fuel during an accident in ground operations. If 7-psi vapor pressure fuel is splashed around and ignited, nothing much is likely to be left of the airplane or the passengers. If safety fuel were used, the fire, if any, is likely to be much less intense, and in many cases time would be available to remove passengers and bring extinguishing equipment to the airplane. It was mentioned above that gasoline itself does not readily ignite on hot surfaces, but that lubricating oil does. From this it has sometimes been inferred that the lubricating oil actually constitutes the greater hazard. This appears justified only if it can be shown that the lubricating oil actually is the source of ignition in most cases. There are many other potential sources of ignition also: all the electrical circuits, hydraulic fluids, fabric and plastics, wood and paper, even the matches in the passengers' pockets, all of which ignite as readily as lubricating oil. Therefore, even if lubricating oils can be made less ignitible, it will be much more satisfactory also to lower the vapor pressure of the fuel. This may not eliminate all fires, but will often eliminate their catastrophical character.

Considering the picture as a whole, it is our opinion that safety fuels indeed offer much added safety where it is most needed and increase the hazard only where it can be controlled.

■ Vapor Lock and Evaporation Losses

It is evident from the previous discussion that with fuels of current volatility, conditions for fuel boiling in either the fuel tank or in the fuel lines may readily occur under normal flight operation. Since for each volume of fuel that is evaporated, approximately 200 volumes of vapor are formed, it is not hard to understand that the fuel system will vapor lock if boiling is allowed to occur in any of its parts. Another factor which aggravates vapor lock is the dissolved air, which is also released under the reduced pressures at altitude. The amounts of vapor formed by aviation gasoline and safety fuel are compared in Fig. 14, which illustrates the advantage of safety fuel in this respect.

A consequence of fuel boiling in the tank is a very undesirable loss of fuel. Even before the boiling altitude has quite been reached a significant amount of fuel vapor will be carried away with the release of dissolved air and breathing of the tanks. The actual boiling altitude for conventional fuels under average conditions is reached at 18,000-20,000 ft. From there on up the loss will increase as altitude increases further. Hence, in a single climb to high altitude, up to 25% of the fuel may boil away. In large planes, this may represent nearly 5000 lb of weight. That much in payload has been lost merely to provide easy starting and good manifold distribution during warmups, takeoff, and climb. Certainly that is not economical.

In the conventional engine the way out will lie in the use of heavier fuels of the safety fuel type. Fuels much above the 300-400 F boiling range do not vaporize satisfactorily in the cylinder and are difficult to handle at low temperatures. The problems of starting and fuel distribution can be eliminated by the use of individual fuel distribution to each cylinder, as already mentioned.

In the gas turbine powered engine relatively heavy fuels are already in use, and problems of vapor lock and altitude fuel loss are thereby avoided.

■ Freezing Point

Another fuel property of importance is the freezing point. The fuel must, of course, be liquid when it is loaded into the airplane tanks. There it may be cooled further by the atmospheric air, but in the case of tanks buried inside the wings or fuselage this effect will be small, because of the small surface to volume ratio, good insulation, and the high specific heat of the fuel. In the carburetor engine the fuel flowing from the tank to the carburetor may drop in temperature. This will be true when the carburetor is cooled by a high velocity air stream whose temperature is lowered some 40 or 50 F below that of the outside air by the evaporation of the fuel. In some cases the fuel is separated from this air stream by a thin aluminum wall only, and it is obvious that the freezing point must be low enough to allow for cooling of the fuel below tank temperature.

In the gas turbine engine the fuel meets heated rather than cooled air, and, provided fuel lines are adequately insulated there is no reason why fuel temperature should anywhere be lower than in the tank. Thus, it may be possible to tolerate a higher fuel freezing point.

■ General Considerations on Fuel Selection

Fuel cost may constitute 10% or more of the total expense of an airline, and will, therefore, influence the direction of future developments. It would be most desirable to know what fuels will cost, and particularly what the relative price ratio between various grades will be. Unfortunately, a price cannot be determined for any one product before the supply and demand position is known, and in addition the market price for every other oil product made from the same barrel of crude. Price predictions are, therefore, as uncertain as predictions on the number of airplanes or on the type of engines they will use.

But it is certain that under normal conditions no fuel will be offered at a price lower than that which it can command in some other field. How the various boiling ranges of the crude oil were used in a typical prewar year is shown in Fig. 15. Conventional aviation gasolines boiling in the 100-300 F range could make outstanding motor car fuel, and will, therefore, cost more than normal motor fuels. Fuels boiling in the 350-700 F range make premium diesel fuels or excellent cracking stocks for eventual use in premium motor fuels. If the gas turbine engine can be developed to use any commercial fuel in the gasoline, kerosene, or diesel fuel range, it will be economically sound to look for the range that is least attractive for other uses. That will probably be the 300-400 F range which, generally speaking, makes poor quality gasoline and is not very

attractive for cracking purposes. However, technological advances and probable postwar changes in the relative demand for the various types of fuels make long-term predictions here also uncertain.

In any case it is well to realize that restrictions with respect to fuel composition or properties usually carry with them a penalty in cost. It pays, therefore, to keep in mind that fuel properties and engine design factors are interdependent – what one cannot economically achieve the other frequently can. Thus, fuel specifications should not be victimized to offset design mistakes, nor of course should a good design be jeopardized by bad fuel. With close cooperation of the oil and aircraft industries it will be possible to develop the best combination of both.

Induction Heat-Treatment

continued from page 54

three parts can be reduced to one, and at the same time the overall diameter of the part can be conveniently reduced. This produces a superior product at a considerable reduction of production work and manufacturing cost. (See Fig. 16.)

■ Bearing Race Application

Where the bearing race is to be pressed into a hub or other part in which a recess has been bored to receive the bearing race, fretting or galling may be reduced to a marked degree by internal heat-treatment of the receding part, followed by precision machining to the tolerances required for the proper fitting of the bearing race into the desired location. (See Fig. 17.)

■ Axle Ends

The wear of the king pin in the axle end is frequently a problem. This wear can be reduced greatly by heat-treating the bore to some relatively high hardness, thus confining the wear substantially to the pin – a replaceable part.

■ Internal Gears

The heat-treatment of gears by induction is too well established to need any considerable discussion. Improved methods of heat-treating internal gears by induction are well advanced, and contour hardening of the gear, together with the trapping of the required type of stress to resist spalling or fatigue, can be accomplished.

■ Internal Splines

Where increased strength and wear resistance are desired, splines may be internally heat-treated. (See Fig. 18.)

■ Grease Retainer Recesses

Where grease retainer recesses are retained within the diameter, frequent wear takes place due to dust. This may be reduced by hardening and changing the metallurgical structure of the part.

■ Hydraulic Cylinders

Precision cylinders of great strength and low weight may be produced through the use of internal heat-treating; or the cylinders may be provided with a very hard case in the bore to resist damage when foreign material, such as abrasives and the like, accidentally enter the oil stream. We induction heat-treat all of our internal cylinders to prevent damage due to foreign bodies, as any tendency to gall or fret is confined to the softer part of the piston and not to the cylinder, the cost of which greatly exceeds that of a piston. (See Fig. 19.)

These applications are adaptable particularly to dump trucks, tractors, bulldozers, road scrapers, and the like.

In such applications cylinders which have been heat-treated throughout to extremely high physicals may then be bore hardened to produce a hard surface resistant to galling and wear. Such cylinders will be made usually of steel but certain applications may be of cast iron.

In approaching these engineering problems many pertinent advantages as to product design and production will be uncovered if the design is studied from the angle of integral heat-treatment, either for hard bearing surfaces or for locally improved physicals. For instance, a localized heat-treat may be applied to counteract the stresses of a shrink fit, such as in the assembly of a railroad car axle and wheel. Certainly, we have found that careful study of a product both by the design engineer and by the induction engineer, usually working together, can prove highly profitable both in improving the quality of a product and in controlling the cost of production.

We discovered some years ago that the hardened structure which we achieve in our method of hardening internal diameters has considerable machinability, the machinability of hardened cast iron being greater than that of hardened steel. We have found that cast-iron cylinders when inductively bore treated can be drilled, and that there is very little danger of spalling on the internal surface when the drill breaks through. Such a hardened cylinder has just recently been produced for one of our large manufacturers for use in an air compressor. We have found it is not practical to heat-treat the bore of a cylinder which has holes in it, such as ports or the like, except between such openings.

In closing, may I say that it is not the source of high-frequency energy that controls the performance of induction heat-treatment. This source can be electronic equipment, spark-gap equipment, or rotating equipment. It is not the heating device alone, although this heating device must be designed with great care and must be combined with means for rotation, oscillation, and general motivation, and all of the accessories. It is not the quenching device, nor the coolant, be it air, water, or oil. It is the judicial combination of all these components into a well-balanced industrial application that assures final success.

There will be many applications of heat-treatment in years to come which are unthought of today, many of which may be of more importance than anything we have with us at present, but the application of internal induction to the automotive industry will help you improve your product and, at the same time, reduce its manufacturing cost.

Effect of ENGINE EXHAUST PRESSURE on

by

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RECENT improvements in the efficiency of centrifugal compressors and exhaust turbines have stimulated interest in the possibility of the compressor-engine-turbine combination for aircraft, in which the turbine is geared to the engine crankshaft.[1, 2, 3] The object of the work reported in this paper was to determine the effect of variations in the engine exhaust pressure on the performance of such units at various altitudes.

The power absorbed by a compressor and the power delivered by a turbine are readily predicted, provided the initial conditions and the pressure ratios are known, and the efficiencies can be estimated. On the other hand, predictions of reciprocating-engine performance are much more difficult since many more variables enter the problem. One of the problems concerning which there has been a scarcity of information is that of the effect of exhaust pressure on indicated mean effective pressure when the exhaust pressure is carried to levels considerably higher than one atmosphere. Also, little published information is available on the temperatures of exhaust gases from reciprocating engines running under various operating conditions. Information in both these fields is necessary before accurate estimates of the performance of compressor-engine-turbine combinations can be made.

The experimental part of this investigation consisted of measurements of indicated mean pressure, fuel consumption, and exhaust temperature on a liquid-cooled aircraft-engine cylinder at various inlet and exhaust pressures. The results of these tests were used in computing the overall performance of compressor-engine-turbine combinations at various altitudes and with various pressures in the inlet and exhaust systems of the engine.

In addition to the computations based on engine tests, a series of computations was made based on the thermodynamic charts for fuel-air mixtures at equilibrium previously published.[4, 5] These computations were made for conditions corresponding to those covered for the compressor-engine-turbine combination.

■ Experimental Work

The experimental work on which this paper is based[6] was conducted in the Sloan Laboratories at the Massachusetts Institute of Technology.

The engine used for the experimental work consisted of a Lycoming liquid-cooled aircraft-type cylinder mounted on a special, single-cylinder crankcase. The cylinder had a bore of 5.25 in., a stroke of 6.25 in., and a compression ratio of 6.0. All runs were made at a piston speed of 3000 fpm (2844 rpm), with a fuel-air ratio of 0.08. Other constants used during the tests are given in Appendix I.

Indicated mean effective pressure was measured by means of the M.I.T. balanced-pressure indicator.[7]

Exhaust temperatures were measured by means of thermocouples located in a special, double-jacketed receiver, shown diagrammatically in Fig. 1. The object of the baffling was to reduce the exhaust gas velocity substantially to zero at the thermocouples and to ensure thorough mixing of the gases before their temperature was measured. The object of the jacketing was to bring the inner chamber wall to the exhaust gas temperature in order to avoid errors due to radiation. It was hoped that these devices would be so effective that all three thermocouples would give the same reading. Unfortunately, this was not the case, and the readings of the three thermocouples differed by amounts up to 100 F. Thermocouple No. 3 (Fig. 1) always ran at the highest temperature, couple No. 1 at the lowest, and No. 2 at a temperature intermediate between the two. Most of the time the difference between

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[1] See SAE Transactions, Vol. 53, June, 1945, pp. 345-351: "Turbine Compounding with Piston Engine," by C. F. Bachle.

[2] See *Mechanical Engineering*, Vol. 64, November, 1942, pp. 779-783: "Supercharging of Two-Stroke Diesel Engines," by F. Oederlin.

[3] See Transactions of Institute of Marine Engineers, Vol. 51, 1939, pp. 139-154 + (disc.) 154-157: "Thermodynamics of New Type of Marine Machinery: Combustion Engines with Pneumatic Power Transmission," by H. G. Hammar and E. Johansson.

[4] See SAE Transactions, Vol. 31, October, 1936, pp. 409-424: "Thermodynamic Properties of Working Fluid in Internal-Combustion Engines," by R. L. Hershey, J. E. Eberhardt, and H. C. Hottel.

[5] See "Internal-Combustion Engine," by C. F. Taylor and E. S. Taylor. International Textbook Co., Scranton, Pa., 1938.

[6] See "Effect of High Exhaust Pressures on Engine Performance and Availability of Energy in Exhaust Gases at High Pressures," by W. E. Kenna, C. Antoniak, and K. B. McCutcheon. Thesis for M. S., M.I.T., 1944.

[7] See *Mechanical Engineering*, Vol. 55, March, 1933, pp. 169-171: "New High-Speed Engine Indicator," by E. S. Taylor and C. S. Draper.

[8] See NACA Technical Report No. 218 (1925), "Standard Atmosphere – Tables and Data," by W. S. Diehl.

[9] See "Effect of Engine Exhaust Pressure on Performance of Compressor-Engine-Turbine Combination," by H. Yegin. Thesis for M. S., M.I.T., 1944.

on Performance of COMPRESSOR-ENGINE-TURBINE UNITS

the first and last thermocouple readings was less than 60 F. The reading of thermocouple No. 3 was taken as the basis for calculating the performance of an exhaust-driven turbine.

No overheating due to high exhaust pressures or other evidence of engine distress was noted during the tests, and the cylinder, piston, and valves were in good condition after the tests were completed. The highest inlet manifold pressure actually used was 50 in. of mercury absolute, at which point there was no detonation with 100-octane aviation fuel. The highest exhaust pressure used on test was 60 in. of mercury absolute.

Table I gives principal engine test results.

■ Calculation of Performance

The arrangement on which calculations for the compressor-engine-turbine unit were based is shown diagrammatically in Fig. 2. The principal assumptions made were as follows:

Compressor efficiency	0.85
Turbine efficiency	0.90
Temperature of gases entering turbine	From curves, Fig. 3
Atmospheric conditions	Standard[8]
Ram pressure at compressor intake, in. Hg	1
Engine indicated mean pressure	From Fig. 4
Engine friction	Estimated from tests on multicylinder engines, Fig. 5

Although tests were not made above 50 in. absolute inlet pressure and 60 in. absolute exhaust pressure, the results were extrapolated to include 60 in. inlet and 80 in. exhaust pressure in the calculations. Test results were corrected to a compression ratio of 6.5 and an engine inlet temperature of 586 R in order that they should be comparable with the theoretical results. Other assumptions and sample calculations are given in Appendix II. A few calculations were also made assuming an efficiency of 0.70 for both compressor and turbine.

The assumption of constant efficiencies for compressor and turbine would not be valid for specific units used over a wide range of operating conditions. The plots, therefore, constitute envelope curves showing performance with compressor and turbine units specifically designed for each set of operating conditions.

■ Calculations for Theoretical Cycles

The theoretical curves included herewith are based on the work of Lt. H. Yegin of the Turkish Republic.[9] The

REPORTED here are the results of work carried on to determine the effect of variations in engine exhaust pressure on the performance at various altitudes of compressor-engine-turbine combinations for aircraft, in which the turbine is geared to the engine crankshaft.

Measurements were made of indicated mean effective pressure, fuel consumption, and exhaust temperature on a liquid-cooled aircraft-engine cylinder at various inlet and exhaust pressures.

The author concludes that this combination unit can give high specific output and low specific fuel consumption, particularly at high altitudes. He also determines the ratio of engine exhaust pressure to engine inlet pressure required for maximum net output, as well as for lowest net fuel consumption. He found further that high exhaust pressures did not appear to impose undue mechanical or thermal stresses on the engine.

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theoretical cycles were calculated by means of thermodynamic charts[4, 5] for a mixture of 0.0782 lb of octane vapor per lb of air at equilibrium. Reversible adiabatic compression and expansion processes were assumed, with no pressure losses due to flow, no heat transfer at any point in the system except in the aftercooler, and no engine friction losses.

The following constants were used in the calculations of theoretical cycles:

Engine compression ratio	6.5
Atmospheric conditions	Standard[8]
Ram pressure at compressor inlet, in. Hg	1
Temperature of mixture entering engine, R	586

The method of calculation follows the methods suggested in the references[4, 5]. This method is described in detail in Appendix III, which also contains sample calculations.

■ Results

Fig. 3 shows the exhaust temperature measurements at thermocouple No. 3 of Fig. 1. These measurements show considerable scatter and inconsistency. For purposes of computation the straight lines were used. These are an attempt to establish averages for the exhaust temperatures as a function of the engine inlet and exhaust pressure.

Fig. 4 shows engine indicated mean effective pressure divided by inlet pressure as a function of exhaust pressure/inlet pressure. The correlation was so good on this basis that the single curve shown in the figure was used for the computations involving the actual engine. For purposes of comparison, a similar curve made with an aircooled cylinder[10] is shown in Fig. 3. The greater slope of this curve is attributable to the large valve overlap used (85 deg as contrasted with only 40 deg overlap for the Lycoming cylinder). As would be expected, large valve overlap is detrimental to good performance when exhaust pressures exceed inlet pressures.

Fig. 6 shows net mean effective pressure of the unit as a function of exhaust pressure, inlet pressure, and altitude for both the compressor-engine-turbine combination and for comparable theoretical cycles. The net mean effective pressure is defined as follows:

I. For the compressor-engine-turbine curves:

Net mep = imep − fmep − smep + tmep

where:

Imep is taken from Fig. 4

Fmep is taken from Fig. 5

$$\text{Smep} = \frac{2 \times \text{compressor work per unit time}}{\text{engine displacement per unit time}}$$

$$\text{Tmep} = \frac{2 \times \text{turbine work per unit time}}{\text{engine displacement per unit time}}$$

II. For the theoretical curves:

Net mep = imep − smep + tmep

where:

Imep is calculated from the thermodynamic charts at fuel-air ratio 0.0782

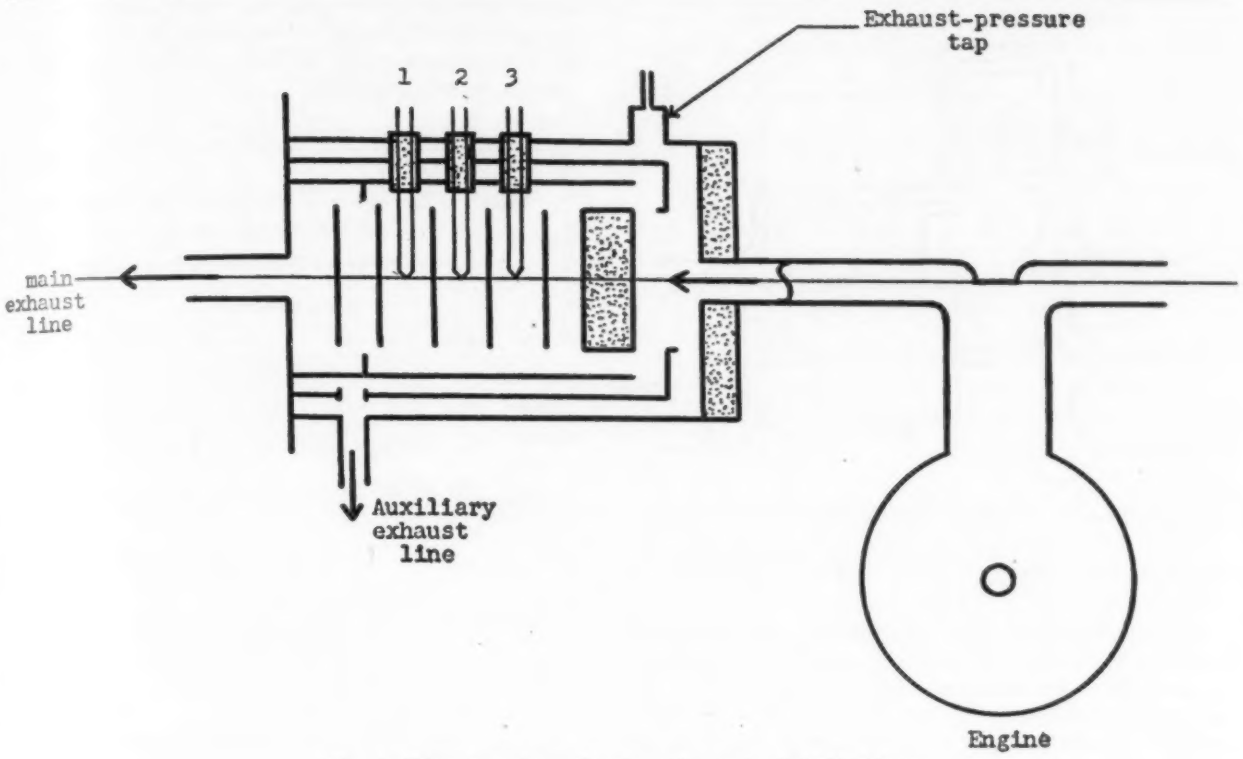
[10] See "Single-Cylinder Investigation of Effect of Varying Exhaust Back Pressure on Engine Performance," by H. E. Bek. Report MR-726, Ranger Aircraft Engines, Division of Fairchild Engine & Airplane Corp., October, 1944.

Table 1 — Complete Engine

P_m	P_i	Run No.	Atmospheric Pressure, in. Hg	B. L., in. Hg	Bmep psi	Bhp	I_{aa}, amp	Rotameter Reading	W_F, lb fuel	P_i, in. Hg	H, in. alcohol	T, F	W_A, lb air	Fuel-Air Ratio	Spark Advance, deg	h_i, in. Hg	h_e, in. Hg	T_i, F	ϵ
30	30	5	29.66	14.6	95.20	47.0	0.13	17.00	30.75	1.21	14.55	80	385	0.0799	28	0.34	0.34	147	0.867
30	30	7	30.04	15.05	97.79	48.3	0.15	17.20	31.00	0.82	14.80	79	388	0.0799	28	−0.04	−0.04	140	0.885
30	40	10	30.04	13.25	86.04	42.5	0.15	15.45	29.10	0.86	13.25	80	366	0.0795	28	−0.03	9.96	140	0.815
30	50	12	30.04	11.28	73.23	36.2	0.12	13.90	27.20	0.65	11.50	80	341	0.0798	28	−0.04	19.96	140	0.759
30	30	1	30.40	14.70	95.49	47.1	0.15	17.10	30.80	0.36	14.50	75	385	0.0800	28	−0.42	−0.40	140	0.855
30	40	2	30.40	13.45	87.34	43.1	0.14	15.75	29.35	0.38	13.25	76	368	0.07975	28	−0.40	9.60	140	0.8175
30	50	3	30.40	11.85	76.94	38.0	0.12	14.55	28.00	0.26	11.85	78	346	0.0810	28	−0.40	19.70	140	0.770
30	60	5	30.40	9.85	64.03	31.7	0.10	13.10	26.20	0.20	10.40	79	324	0.0808	28	−0.40	30.40	140	0.722
40	30	7	30.40	22.40	145.56	71.9	0.19	29.00	43.40	10.75	21.45	73	543	0.0800	28	9.55	−0.40	140	0.906
40	40	8	30.40	21.50	139.66	69.0	0.19	27.70	41.80	10.80	19.85	73	523	0.0800	28	9.60	9.60	140	0.869
40	50	3	29.71	19.85	128.95	63.7	0.17	25.75	39.80	11.35	18.15	70	500	0.0796	28	10.29	20.29	140	0.834
40	60	4	29.71	18.00	116.94	57.8	0.15	23.80	37.80	11.20	16.40	71	474	0.0798	28	10.29	30.29	140	0.791
50	60	2	30.47	26.35	171.00	84.5	0.20	35.55	50.65	21.25	23.00	76	628	0.0807	28	19.95	29.95	140	0.8378
50	50	5	30.47	28.10	182.50	90.2	0.22	37.30	52.60	21.40	25.00	76	651	0.0808	28	19.95	19.95	140	0.869
50	40	7	30.47	29.90	194.00	95.8	0.22	38.45	53.90	21.60	27.00	78	679	0.0795	28	19.95	9.35	140	0.904
50	30	9	30.47	30.60	198.50	98.0	0.23	40.10	55.75	21.65	28.50	78	697	0.0810	28	19.95	−0.05	140	0.930

Note: $P_m = P_i$ = Inlet pipe pressure.

* From indicator cards, not including pumping cycle.



■ Fig. 1 – Exhaust receiver – thermocouples were of shielded type

$$\text{Smep} = \frac{\text{reversible adiabatic work of compression from atmosphere to engine inlet pressure, per lb air}}{\text{cylinder displaced volume per lb air}}$$

$$\text{Tmep} = \frac{\text{reversible adiabatic work of expansion from engine exhaust pressure and temperature to atmospheric pressure, per lb air}}{\text{cylinder displaced volume per lb air}}$$

Both theoretical and experimental curves are given in Fig. 6. It is interesting to note that both sets of curves show maximum mep very near the point where the engine exhaust and inlet pressures are equal.

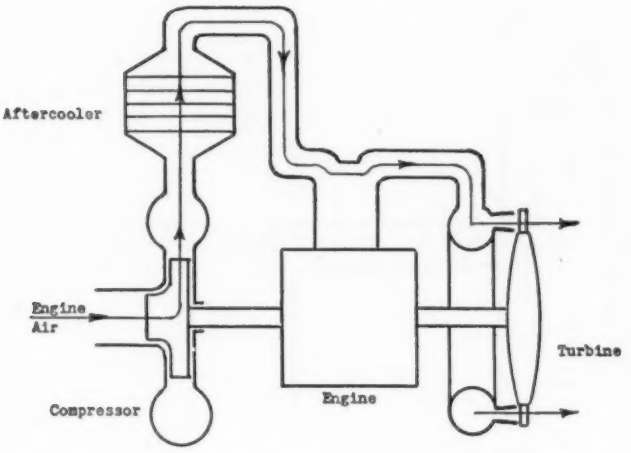
Figs. 7 and 8 show the mean effective pressures of the separate units – engine, compressor, and turbine – at 15,000- and 30,000-ft altitude. The engine mep is much higher than that of the turbine under all circumstances. The difference between turbine and compressor mep shows the sacrifice in output which would be involved if the turbine were connected to the compressor only, as in the case of current turbosupercharger installations. This sacrifice would decrease with decreasing efficiencies of turbine and compressor.

Table 2 compares the performance of the compressor-engine-turbine combination at the maximum-mep conditions with that of the engine exhausting to atmospheric pressure, with a gear-driven compressor of 0.85 efficiency. The table may be slightly unfair to the compressor-engine combination in that up to 10% additional power is obtainable from exhaust thrust in a fast airplane when an exhaust turbine is not used.

Fig. 9 shows net specific fuel consumption, that is, the fuel consumed per horsepower hour of the whole system. Lowest specific fuel consumption occurs at a higher value of the exhaust pressure than that for maximum mep in every case.

Test Results

Imep,* psi	T_1	T_2	T_3	T_4	P_{oil}, psi	T_{oil}, F	T_{cyl}, F	P_{cyl}, psi
—	33.0	33.8	34.3	8.25	70	170	183	—
	1459	1494	1515	1625				
145	34.1	34.8	35.3	8.5	—	150	180	—
	1506	1538	1560	1663				
135	32.6	33.6	34.2	8.67	64	150	180	56
	1441	1486	1507	1690				
125	30.6	32.0	32.9	9.05	64	155	180	56
	1355	1415	1455	1748				
145	33.8	34.6	35.2	8.40	56	150	180	—
	1493	1529	1554	1647				
135	31.6	32.7	33.4	8.55	57	150	180	—
	1398	1446	1476	1670				
125	29.7	30.8	31.5	8.67	60	150	180	—
	1317	1364	1393	1690				
115	26.3	27.6	28.5	8.80	64	150	180	—
	1171	1228	1266	1710				
195	36.7	37.4	38.1	8.62	58	150	180	—
	1623	1654	1684	1682				
188	34.4	35.3	35.7	8.42	58	150	180	—
	1520	1564	1577	1850				
178	34.1	34.9	35.3	—	56	150	180	—
	1507	1542	1559	—				
166	33.5	34.3	34.6	—	55	150	180	—
	1480	1515	1528	—				
228	35.3	35.8	36.2	38.50	71	76	—	62
	1559	1582	1600	1702				
239	35.6	36.0	36.4	37.60	72	76	—	62
	1572	1590	1610	1662				
249	35.5	35.9	36.4	37.40	70	78	—	62
	1568	1586	1610	1654				
255	36.0	36.5	37.0	37.40	70	78	—	62
	1590	1614	1636	1654				



■ Fig. 2 – Diagram of assumed compressor-engine-turbine system

Fig. 10 shows the effect of lowering compressor and turbine efficiencies on the performance of the compressor-engine-turbine combination. Lowering the compressor and turbine efficiencies not only lowers the net output, but also reduces the ratio exhaust pressure/inlet pressure at which maximum mep occurs. Lowest specific fuel consumption, however, remains at nearly the same exhaust pressure.

■ Conclusions

1. Compared with the conventional aircraft engine, high specific output and low specific fuel consumption can be obtained with the engine-compressor-turbine combination, particularly at high altitudes.

2. Under the conditions assumed in this work, the ratio of engine exhaust pressure to engine inlet pressure should be in the range 0.70 to 1.1 for maximum net output. For lowest net fuel consumption, the ratio of exhaust pressure to inlet pressure should be in the range 1.4 to 2.0.

3. Theoretical cycles show the same general trends, with varying exhaust pressure, as the compressor-engine-turbine combination. However, the ratios of exhaust to inlet pressure for maximum output and for lowest specific consumption are somewhat higher.

4. In the limited amount of running involved in the engine performance tests, there was no indication that high exhaust pressures imposed undue mechanical or thermal stresses on the engine.

■ Suggested Further Work

The work reported herewith has been confined to fuel-air ratios in the neighborhood of 0.08, and to engine compression ratios of 6.0-6.5. It would be of interest to extend such work to other fuel-air ratios and compression ratios.

In this work it has been assumed that all of the air from the compressor passes through the engine and turbine and that fuel is burned only in the engine. It would be of interest to explore combinations where some of the air is bypassed around the engine to the turbine. In such combinations the effect of introducing and burning fuel between engine and turbine could be explored. It is believed that sufficient data on engine performance are presented herein so that computations for various systems in this category could be made with the assurance of reasonably good accuracy.

Table 2 — Comparison of Maximum Values of Net Mep with and without Turbine

Fuel-air ratio = 0.08, compression ratio = 6.5

Altitude, ft	Inlet Pressure, in. Hg	Maximum Net Mep: Compressor-Engine*	Maximum Net Mep: Compressor-Engine-Turbine	Isfc at Max Mep: Compressor-Engine	Isfc at Max Mep: Compressor-Engine-Turbine
30,000	60	254 at p_e=8.9 in.	362 at p_e=50 in.	0.518	0.378
	40	169 at p_e=9.9 in.	229 at p_e=33 in.	0.568	0.410
15,000	60	270 at p_e=16.9 in.	338 at p_e=55 in.	0.530	0.395
	40	176 at p_e=16.9 in.	202 at p_e=38 in.	0.540	0.430
Sea level	60	281 at p_e=29.9 in.	313 at p_e=63 in.	0.505	0.420
	40	181 at p_e=29.9 in.	182 at p_e=43 in.	0.520	0.477

p_e = Engine exhaust pressure

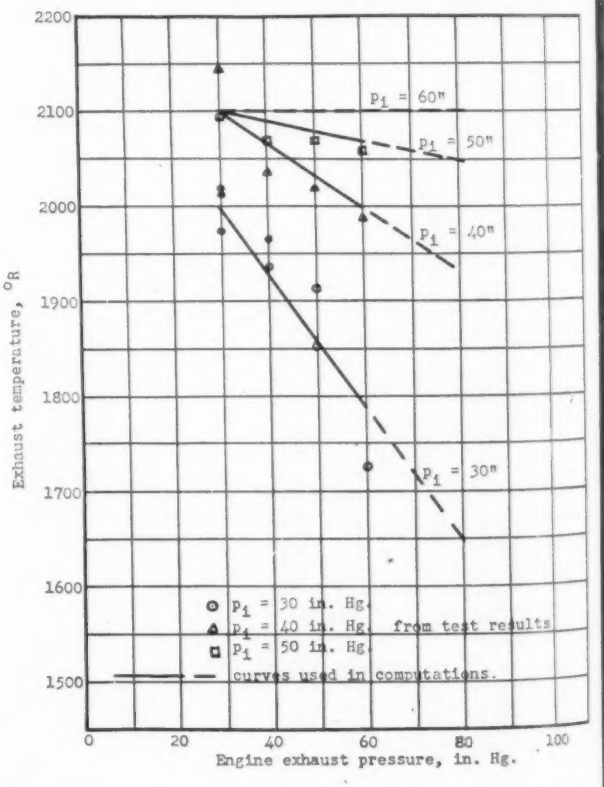
* Engine assumed to exhaust to atmospheric pressure. Compressor is gear driven. Compressor efficiency = 0.85, turbine efficiency = 0.90.

APPENDIX I

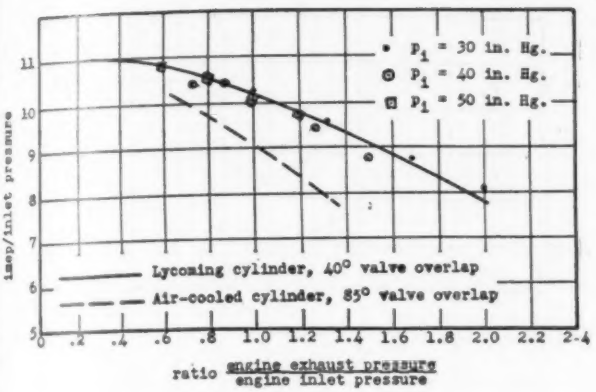
Engine Tests

Engine: Single-cylinder Lycoming, 5¼-in. bore, 6¼-in. stroke, compression ratio = 6.0

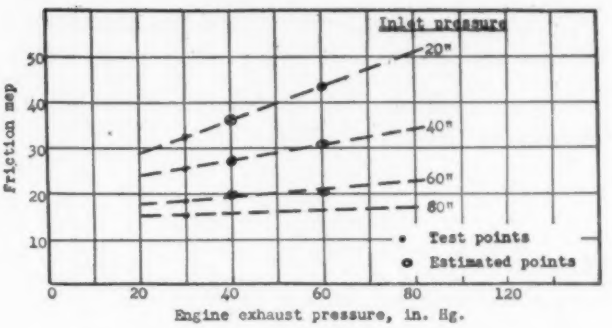
Valve timing: Inlet opens, deg BTC.....20
Inlet closes, deg ABC.....56
Exhaust opens, deg BBC.....64
Exhaust closes, deg ATC.....20



■ Fig. 3 – Exhaust gas temperatures at thermocouple No. 3, Fig. 1


imep/inlet pressure
P_i = 30 in. Hg.
P_i = 40 in. Hg.
P_i = 50 in. Hg.
Lycoming cylinder, 40° valve overlap
Air-cooled cylinder, 85° valve overlap
ratio engine exhaust pressure / engine inlet pressure

■ Fig. 4 – Effect of exhaust pressure on indicated mean effective pressure



■ Fig. 5 – Estimated friction mep for full-scale aircraft engines

Constant test conditions:

Rpm.....2884 (3000 fpm piston speed)

Fuel-air ratio.....0.08 = 0.001

Inlet mixture temperature, F...140 (600 R)

Spark advance, deg.....28

Fuel grade.....100/130 aviation gasoline

Oil pressure, psi.....55-70

Inlet oil temperature, F.....150

Cooling water in, F.....175

Cooling water out, F.....185

Indicated mean effective pressures were obtained by means of the M.I.T. balanced-diaphragm indicator.[7]

Complete test results are given in Table 1.

Methods of correction: In order to place the engine imep on a basis comparable to the calculations of theoretical cycles, it was corrected to a compression ratio of 6.5 and a manifold temperature of 126 F (586 R) as follows:

Corrected imep (Fig. 4) = test imep (from area of indicator diagrams, *not* including pumping cycle) $\times$ $\sqrt{\frac{600}{586} \times \frac{0.526}{0.513}}$, where 0.526 is the air cycle efficiency at 6.5 compression ratio and 0.513 is the air cycle efficiency at 6.0 ratio. The complete correction factor is equal to 1.04.

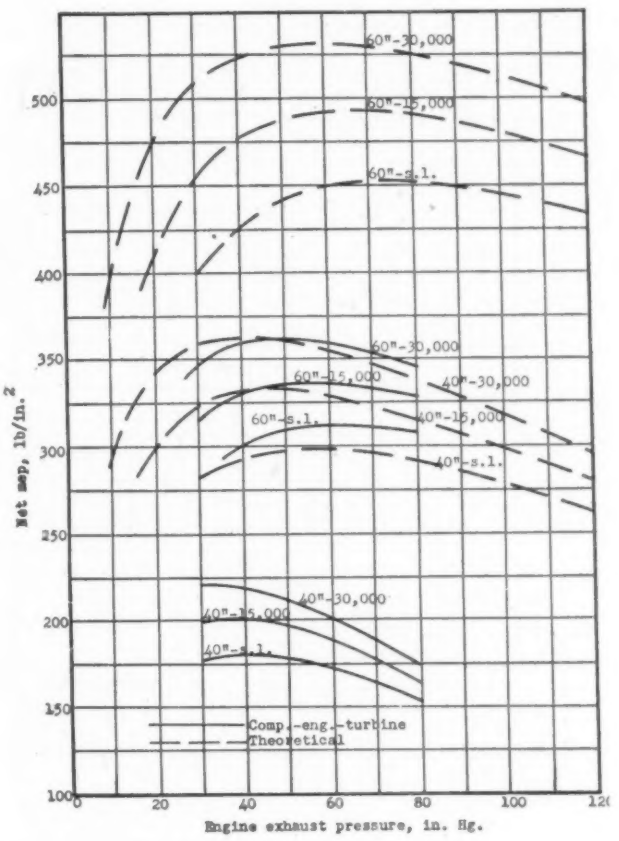
No correction was made on the exhaust gas temperatures (Fig. 3) on account of the assumed increase in compression ratio.

Friction mep (Fig. 5): These curves were constructed from the following assumptions for the friction of a full-scale aircraft engine at 3000 fpm piston speed:

(a) Mechanical friction = 20 psi

(b) Pumping friction mep is as follows:

	p_e		
p_i, in.	30 in.	40 in.	60 in.
20	13	16	24
40	6	7.5	11
60	−1	0	1



■ Fig. 6 – Net mean effective pressure

The foregoing figures are based on such measurements of full-scale engine friction as were available to the author.

Volumetric efficiency (Fig. 11, lower curve): Volumetric efficiency was defined as follows:

$$e = \frac{2W_A}{60\ D\ N\ \rho_i}$$

where:

W_A = Air (measured), lb per hr
D = Piston displacement, cu ft
N = Engine speed, rpm
ρ_i = Density of air in inlet pipe, lb per cu ft

For these tests, $D = 0.0782$, $N = 2882$, $\rho_i = 0.00222\ p_i$ (at 600 R) and therefore:

$$e = 0.0667\ \frac{W_A}{p_i}$$

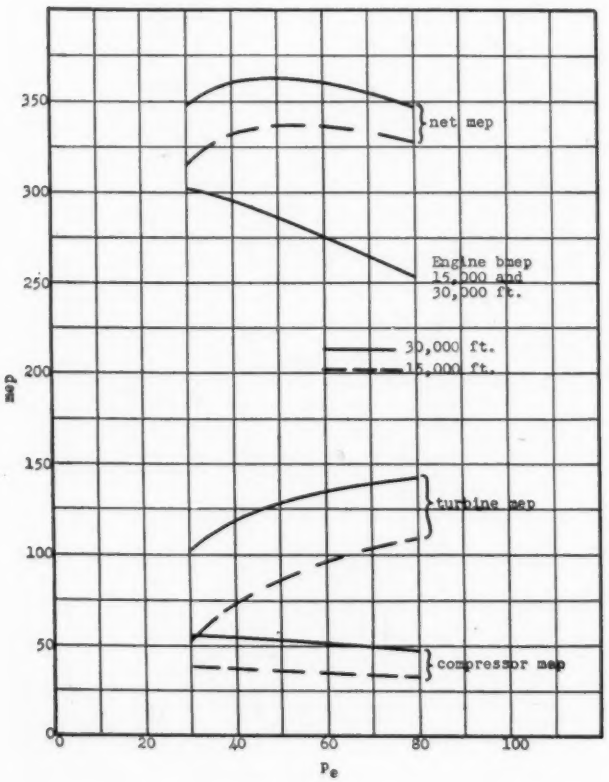
where:

p_i is the inlet pipe pressure, in. of mercury.

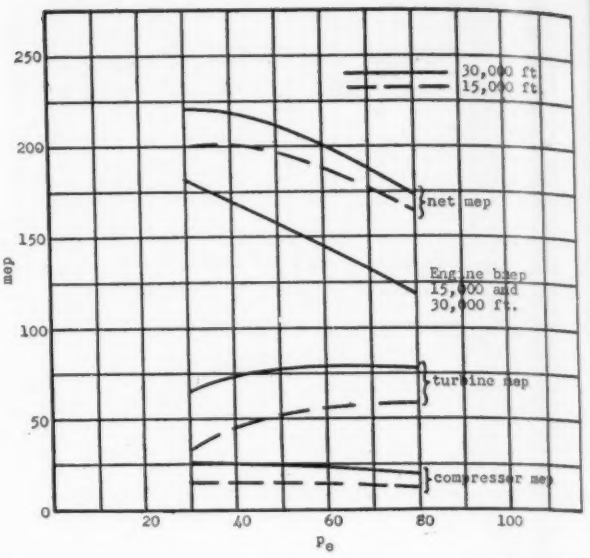
APPENDIX II

Computation of Net Mep and Net Sfc
for Compressor-Engine-Turbine Combination

Atmospheric conditions (from Diehl[8]):



■ Fig. 7 – Mean effective pressure for compressor-engine-turbine combination – 60-in. manifold pressure



■ Fig. 8 – Mean effective pressure for compressor-engine-turbine combination – 40-in. manifold pressure

Altitude, ft	p_a Pressure, in. Hg (abs)	t_a Temperature, F	T_a Temperature, R
Sea level	29.92	60	520
15,000	16.88	5	455
30,000	8.88	−48	412

Compressor (supercharger) inlet conditions (due to 1 in. Hg ram)

Altitude, ft	p_b Pressure, in. Hg (abs)	t_b Temperature, F	T_b Temperature, R
Sea level	30.9	64	524
15,000	17.9	12	472
30,000	9.9	−36	424

Engine inlet conditions:

Inlet manifold pressure, p_i either 40 in. or 60 in. Hg (abs)

Inlet manifold temperature, T_i, 586 R (126 F)
Engine exhaust pressure: p_e, 20-80 in. Hg (abs)
Compressor (or supercharger) mep:

$$\text{Smep} = \frac{778}{144}\ \rho_i\ e\ C_p\ T_b \left[\left(\frac{p_i}{p_b \times 0.98}\right)^{0.283} - 1\right]\frac{1}{\eta_c}$$

where:

C_p = Specific heat of air at constant pressure = 0.24 Btu per lb F

ρ_i = Inlet density (see Appendix I)

e = Volumetric efficiency (from Fig. 11)

η_c = Compressor efficiency

The factor 0.98 is inserted to allow for 2% pressure losses between compressor and engine manifold.

Turbine mep:

$$\text{Tmep} = \frac{778}{144}\,\rho_i\, e\,(h_s - h_e)\,(1 + F)\eta_t$$

where:

h_s is the enthalpy at engine-exhaust pressure p_s and temperature T_s, Btu per lb air

h_e is the enthalpy, Btu per lb air,[11] after reversible adiabatic expansion from exhaust conditions, p_s and T_s, to atmospheric pressure p_a

F is the fuel-air ratio

η_t is the turbine efficiency

Engine imep is taken from test results, Fig. 4.

Friction mep, fmep, is taken from Fig. 5.

Engine bmep = engine imep — fmep.

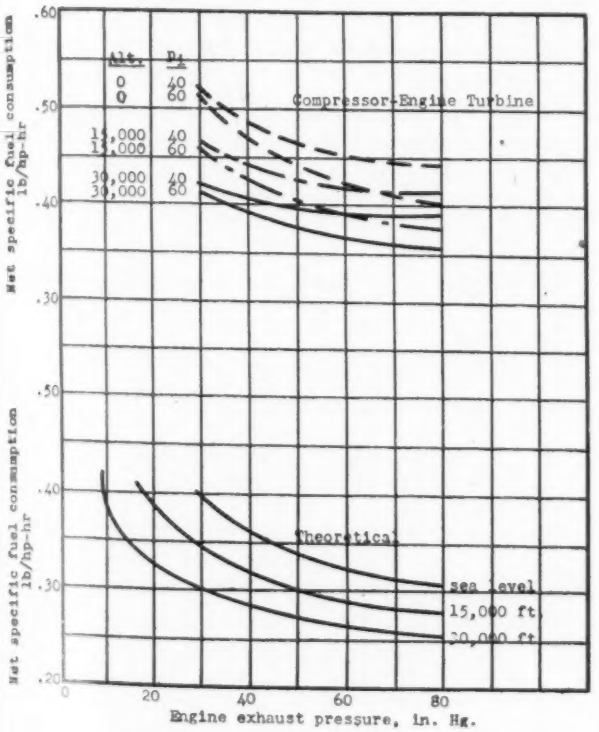
Net mean effective pressure is computed from the relation:

$$\text{Net mep} = \text{imep} - \text{fmep} - \text{smep} + \text{tmep}$$

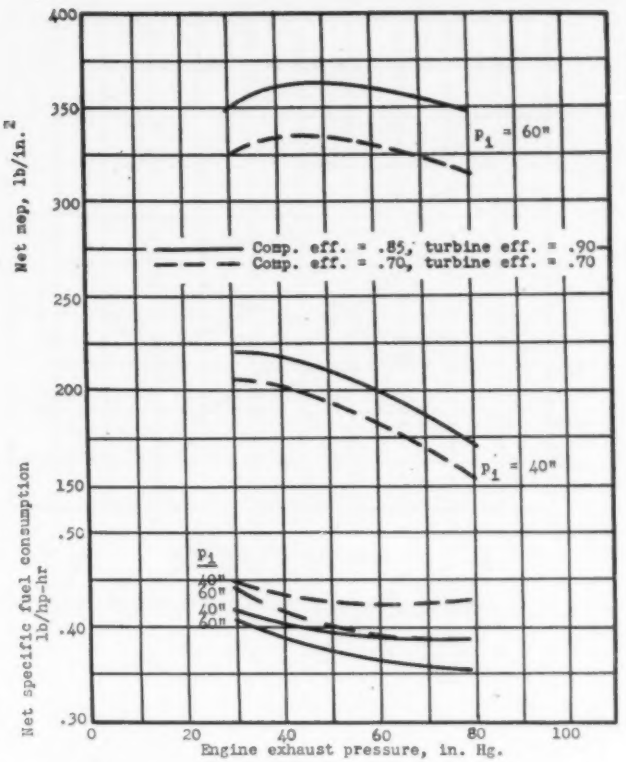
Net specific fuel consumption: The indicated specific fuel consumption of the engine was constant during all tests at 0.444 lb fuel per ihp-hr. The net sfc was, therefore, calculated as follows:

$$\text{Net sfc} = 0.444 \times \frac{\text{imep}}{\text{net mep}}$$

[11] See "Thermodynamic Properties of Air," by J. H. Keenan and J. Kaye, Wiley, New York, 1945.



■ Fig. 9 – Net specific fuel consumption



■ Fig. 10 – Effect of compressor and turbine efficiency on performance of compressor-engine-turbine unit at 30,000 ft

Sample calculation:

Let p_i = 60 in., p_s = 40 in., altitude = 15,000 ft, η_s = 0.85, η_t = 0.90

$$\text{Imep from Fig. 4} = \frac{10.75 \times 60}{2.04} = 316.2 \text{ ps}$$

Fmep from Fig. 5 = 19.5 psi

$\rho_i = 1.33 \times 60/586 = 0.136$

e from Fig. 11 = 0.915

$$\text{Smep} = \left(\frac{778}{144}\right) 0.136 \times 0.915 \times 0.24 \times 472 \left[\left(\frac{60}{17.9 \times 0.98}\right)^{0.283} - 1\right]\frac{1}{0.85} = 37.5 \text{ psi}$$

T_s from Fig. 3 = 2100 R

$$\text{Tmep} = \left(\frac{778}{144}\right) 0.136 \times 0.915 \times (437.4 - 325.9)\, 0.90 \times 1.08 = 72.7 \text{ psi}$$

Net mep = 316.2 − 19.5 − 37.5 + 72.7 = 331.9 psi

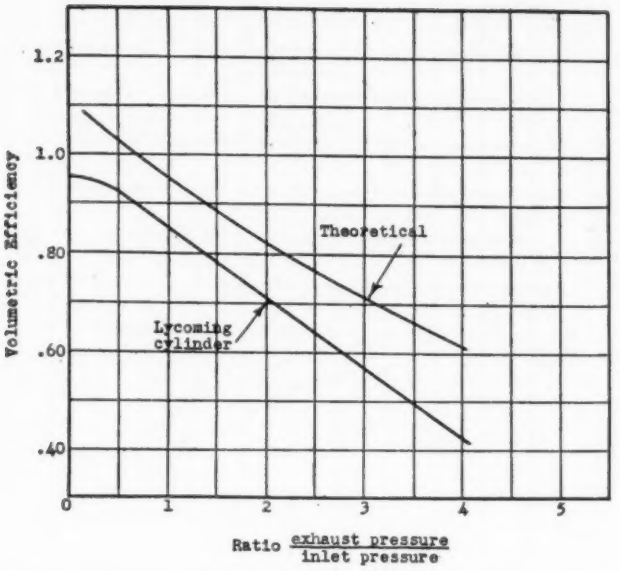
Engine bmep = imep − fmep = 316.2 − 19.5 = 296.7 psi

APPENDIX III

Computation of Net Mep and Net Sfc for Theoretical Cycles

Atmospheric conditions: Same as for compressor-engine-turbine unit (see Appendix II).

Compressor inlet conditions: Same as for compressor-engine-turbine unit (see Appendix II).



■ Fig. 11 – Volumetric efficiency plotted against ratio of exhaust to inlet pressure

Engine inlet conditions:

Inlet manifold pressure, p_i, 40 in. and 60 in. Hg (abs)

Inlet manifold temperature, T_i, 586 R (126 F)

Engine exhaust pressure: p_e, 8.9-120 in. Hg (abs)

In the formulas and computations which follow:

E is the internal energy obtained from Fig. 14 of an article published earlier[4]

E_s is the "sensible" internal energy from Fig. 10 or Fig. 14, of the same article[4]

H and H_s are the enthalpies corresponding to E and E_s

V is the volume of $(1 + F)$ lb of mixture, from Fig. 10 or Fig. 14 of the reference[4]

p is pressure, psi

T is temperature, R

Subscripts refer to points on Fig. 12 of this paper, or to conditions $_i$ in the inlet manifold or $_e$ in the exhaust pipe.

Engine imep is calculated from the thermodynamic charts in accordance with methods suggested in the references[4, 5]. A sample calculation is included in this appendix.

Compressor mep is calculated by the same method as in the case of the compressor-engine-turbine unit (see Appendix II).

Turbine mep is computed from the relation:

$$\text{Tmep} = \left(\frac{778}{144}\right)\frac{e}{V_i}(H_{s_4} - H_{s_5})$$

where:

e = volumetric efficiency of the fuel-air cycle =

$$\frac{(1-f)V_i}{V_1\left(1-\frac{1}{r}\right)}$$

f = Mass fraction of residual gases in cylinder during compression and expansion strokes (see sample calculation)

r = Compression ratio of engine

H_{s_e}, the sensible enthalpy of the gases in the exhaust pipe, is computed from the relations:

$$(H_i - H_e)(1-f) = w$$
$$H_i = H_{s_i} + 1507$$
$$(H_e - H_{s_e}) = (E_e - E_{s_e})$$

w, being the work of the engine cycle in Btu per $(1 + F)$ lb of cylinder contents (see sample calculation)

H_{s_a} = Sensible enthalpy of $(1 + F)$ lb exhaust gases after reversible adiabatic expansion from p_e, T_e, to p_a (atmospheric pressure)

Net mep = imep − smep + tmep

Net sfc $= \dfrac{2545}{19{,}270\eta} \times \dfrac{\text{imep}}{\text{net mep}}$, where η is the efficiency of the engine cycle (see sample calculation following):

Sample Calculation

Altitude, ft	30,000
p_i, in. Hg (abs)	60
p_e, in. Hg (abs)	40

Imep (engine): The theoretical indicator diagram assumed for these calculations is shown in Fig. 12. Processes 1-2, 3-4, 4-5, 6-7, are reversible and adiabatic. 2-3 represents the pressure rise due to combustion to equilibrium. 4-5a represents the pressure drop when the exhaust valve opens. At point 5a the burned gases left in the cylinder after the exhaust valve opens are pushed out of the cylinder at pressure p_e by piston motion. The condition of these gases is represented by reversible adiabatic expansion 4-5 (dash line). At point 6 (end of the exhaust stroke) the exhaust valve closes and the gases are compressed to pressure p_i by an assumed additional motion of the piston. At point 7 the inlet valve opens and the inlet stroke proceeds at pressure p_i to volume V_1. The compression ratio, $r = V_1/V_2 = V_{5a}/V_6$.

In order to construct the engine cycle, it is necessary to assume a temperature for the residual gases in the cylinder at point 7. The assumption will be verified as the calculations proceed, and, if incorrect, a new assumption can be made and the computation repeated. Let T_7 be estimated at 2310 R.

The fraction of residual gas may be approximated by:

$$f \cong \frac{1}{r-1} \times \frac{p_e}{p_i} \times \frac{T_i}{T_7}$$

where $r = 6.5$ compression ratio:

$$f \cong \frac{1}{5.5} \times \frac{40}{60} \times \frac{586}{2310} = 0.0308$$

This value is higher than the exact value, due to the increase in number of molecules after combustion. Six per cent molecular increase is a good average, so let:

$$f = \frac{0.0308}{1.06} = 0.029$$

Inlet process: It is assumed that the fresh charge enters without receiving heat from or giving up heat to the engine parts, and mixes with the residual gases which

were left in the clearance space, the resultant pressure being p_7.

For the process 7-1:

$$H_{a1} = fH_{a7} + (1-f)H_{a7}$$

$$= 0.029(615) + 0.971(55) = 71.2$$

From Fig. 10 of reference 4:

$$T_1 = 644,\ V_1 = 8.39,\ E_{a1} = 25.8$$

For the process 1-2:

$$V_2 = V_1/6.5 = 1.29$$

following line of constant entropy on Fig. 10 of reference 4, from point 1 to V_2 gives:

$$T_2 = 1180,\ p_2 = 361,\ E_{a2} = 143.5$$

For the process 2-3:

$$E_3 = E_{a2} + E_c$$

From Fig. 10 of reference 4,

$$E_c = 1507\ (1-f) + 300f = 1472 \text{ Btu}$$

$$E_3 = 1472 + 143.5 = 1615.5$$

From Fig. 14, reference 4:

$$V_3 = V_2 = 1.29,\ T_3 = 5111,\ p_3 = 1663$$

For the process 3-4:

Following line of constant entropy (Fig. 14 of reference 4) from point 3 to $V_4 = V_1 = 8.39$ gives:

$$T_4 = 3292, p_4 = 162,\ E_4 = 1007$$

Residual gas (process 4-5): When the exhaust valve opens at point 4, the gases which remain in the cylinder expand reversibly and adiabatically to exhaust pressure. Following line of constant entropy from point 4 to $p_e =$ 19.6 psi (40 in. Hg) gives:

$$T_5 = 2107,\ V_5 = 44.7,\ E_5 = 673$$

Fraction of residual gas: The estimated value of f can now be verified, since:

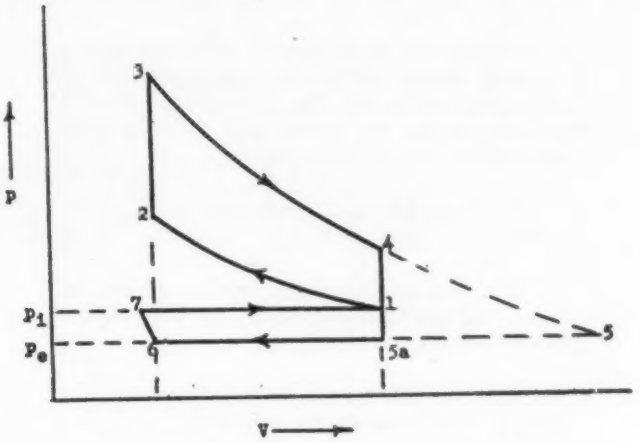
$$= \frac{V_2}{V_5} = \frac{1.29}{44.7} = 0.029$$

which checks with the original estimate.

Compression of residuals (process 6-7): Following constant entropy line (in Fig. 14 of reference 4) from conditions 5 to $p_1 = 29.5$ psi (60 in. Hg) gives:

$$T_7 = 2310,\ E_7 = 725,\ V_7 = 32.6$$

T_7 checks the original estimate and the computation need not be repeated.



■ Fig. 12 – Indicator diagram of theoretical cycle

Work: From the first law of thermodynamics:

$$w = (E_3 - E_4) - (E_{a_2} - E_{a_1})$$

$$-f\,(E_7 - E_6) - \frac{778}{144}\left[p_6(V_1 - V_2) - p_1(V_1 - fV_7)\right] = 505 \text{ Btu}$$

Efficiency of the engine cycle:

$$\eta_i = \frac{505}{1507 \times 0.971} = 0.345$$

Imep of the engine cycle:

$$\text{Imep} = \frac{778}{144}\,\frac{w}{V_1 - V_2} = \frac{778}{144} \times \frac{505}{7.1} = 384 \text{ psi}$$

Volumetric efficiency of the engine:

$$e_v = \frac{(1-f)V_i}{(V_1 - V_2)} = \frac{0.971 \times 7.35}{7.1} = 1.005$$

since $p_i = \dfrac{1}{V_i}$

Compressor mep:

$$\text{Smep} = \frac{778}{144} \times \frac{e}{V_i} \times C_p T_i \left\{\left(\frac{p_i}{0.98\ p_b}\right)^{0.283} - 1\right\}$$

$$= \frac{778}{144} \times \frac{1.005}{7.35} \times 0.24 \times 424 \left\{\left(\frac{60}{0.98 \times 9.9}\right)^{0.283} - 1\right\}$$

$$= 51 \text{ psi}$$

Turbine mep:

$$\text{Tmep} = \frac{778}{144}\,\frac{e}{V_i}\,(H_{a5} - H_{a6})$$

$$H_e = H_i - w/(1-f)$$

$$= 55 + 1507 - 505/0.971 = 1042$$

$H_e - H_{a5} = E_e - E_{a5} = 295$ (from Fig. 14 of reference 4

$H_{a5} = 1042 - 295 = 747$

At $H_{a5} = 747$, $p_e = 40/2.04 = 19.6$ psi

$T_e = 2700$R

Following a line of constant entropy to $p_6 = 8.9/2.04 =$ 4.36 psi gives $H_{a6} = 488$ and:

$$\text{Tmep} = \frac{778}{144} \times \frac{1}{7.35} \times 1.005(747 - 488) = 191 \text{ psi}$$

Net mep $= 384 - 51 + 191 = 524$ psi

$$\text{Net sfc} = \frac{2545}{19{,}270 \times 0.345} \times \frac{384}{524} = 0.282 \text{ lb per hp-hr}$$

Use and Evaluation of

by

O. L. BRANDES and H. H. DONALDSON, JR.

Gulf Research & Development Co.

IN carrying out their purpose of giving here a general survey of the use and evaluation of heavy-duty engine oils, the authors have divided their subject into four parts, each of which gives their answer to a question, namely:

1. What are heavy-duty oils and how are they made?

2. Can the engine performance quality of such oils be predicted from conventional inspection test data?

3. How can oils be properly evaluated as to engine performance characteristics?

4. When is it necessary to use heavy-duty oils, and how should they be used?

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THE past decade has witnessed steadily advancing progress in the development of engines for both heavy-duty gasoline and high-speed, diesel-driven equipment. Paralleling and made necessary by this advance is the similar progress made in the development of remarkably improved engine lubricating oils, commonly referred to as heavy-duty oils. The success of both these advances is well exemplified by the outstanding performance of these modern engines and oils on both the home-front and the battle-fronts.

In preparing this paper on heavy-duty oils, it was believed that a rather general survey of the subject might be of more interest than would a detailed account of any particular phase of the development, use, or evaluation of these oils. Based on this assumption, therefore, the paper is directed toward a general consideration of some of the items which frequently arise in discussions on the subject of heavy-duty oils; namely:

1. What are heavy-duty oils and how are they made?

2. Can the engine performance quality of such oils be predicted from conventional inspection test data?

3. How can oils be properly evaluated as to engine performance characteristics?

4. When is it necessary to use heavy-duty oils, and how should they be used?

I. What Are Heavy-Duty Oils and How Are They Made?

Modern heavy-duty engine oils are generally considered to be oils of the type defined by U. S. Army Specification 2-104B and Bureau of Ships Specification 14-0-13(INT).

Oils qualified under Specification 2-104B are used for the lubrication of engines in all army ground-force equipment. The specification covers three viscosity grades, that is, SAE 10, 30, and 50, and requires acceptable performance in the prescribed gasoline- and diesel-engine qualification tests. A stable pour point requirement for the SAE 10 grade, and foam test limits for the SAE 30 and 50 grades are also included in the specification.

In general, oils conforming with these specifications must:

1. Provide satisfactory engine lubrication even under very severe operating conditions.
2. Keep piston rings free and oil channels clean.
3. Minimize cylinder and ring wear.
4. Not corrode bearings and engine parts.
5. Not cause excessive engine deposits (carbon, lacquer, varnish).
6. Possess satisfactory pour point and antifoam characteristics.

Briefly, heavy-duty oils possessing these properties are made by incorporating in the proper petroleum-base stock (straight mineral oil) suitable proportions of certain chemical agents, commonly referred to as additives, or addition agents, to obtain improved engine oil performance. To develop the proper combination of base oil and additive requires, of course, extensive research and testing in engines.

II. Can Engine Performance Characteristics of Heavy-Duty Oils Be Predicted from Conventional Oil Inspection Tests?

Although the existence of modern heavy-duty engine oils of improved engine performance quality and the history of the development of such oils are quite well known, it is apparently not always clearly recognized that the engine performance quality of oils cannot be predicted by conventional oil inspection tests which are commonly used for specification, control, and identification purposes.

To illustrate this point, engine test data on a number of

[This paper was presented at a meeting of the Pittsburgh Section of the SAE, Pittsburgh, May 9, 1945.]

HEAVY-DUTY ENGINE OILS

oils will be compared with typical inspection tests on the same oils.

Acid or Base Number versus Engine Performance – Fig. 1 shows the total acid number, or, as most often referred to, the neutralization number, of a series of oils plotted against the copper-lead bearing weight loss figures for the same oils in 36-hr Chevrolet engine tests (280 F oil temperature, 200 F jacket coolant temperature, 30-hp load at 3150 rpm – equivalent to road load at 60 mph, Cu-Pb bearings on two con-rods).

The total acid number referred to is that given by ASTM Designation D663-44T, "Acid and Base Numbers of Petroleum Products by Color-Indicator Titration," wherein the total acid number is defined as the number of milligrams of potassium hydroxide required to react with one gram of sample when tested in accordance with the prescribed method.

Some of the new compounded oils when tested by this procedure show alkaline rather than acid characteristics. This alkalinity is expressed as strong base number, and is defined as the number of milligrams of potassium hydroxide equivalent to the acid required to react with one gram of sample. To simplify the comparison, the alkalinity of oil No. 1 in Fig. 1 is designated as a negative acidity rather than as a strong base number.

As recognized in D663-44T, oils may show apparent acidity or alkalinity even though no free acid or alkaline material is present in the oils, for example, the method (D663-44T) states:

"This method of test is designed to indicate in petroleum products and lubricants the presence and concentration of constituents having acid or alkaline characteristics. . . .

"It is not intended to measure an absolute acidic or basic property that can be used to characterize the oil. . . .

"No general relationship between corrosive acids and acid number by any method is known. . . .

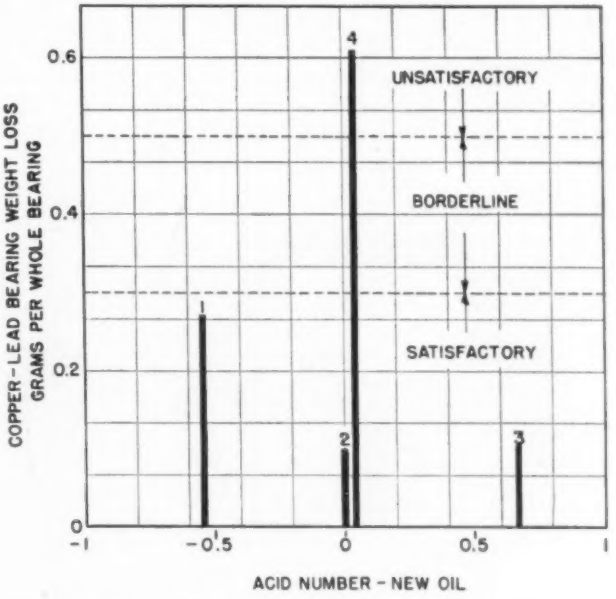
"The compounds present in new and used oils which may consume base include organic and inorganic acids, esters, phenolic compounds, lactones, resins, salts of heavy metals, and addition agents such as inhibitors and detergents. Compounds which cause the oil to be basic to phenolphthalein, and which consume acid, include organic and inorganic bases, amino compounds, soaps, salts of heavy metals, and addition agents such as inhibitors or detergents."

As shown by the data in Fig. 1, a knowledge of the alkaline or acid characteristics of an oil, determined as total acid number (neutralization number) or strong base number gives no information as to the probable engine performance characteristics of the oil. Oils Nos. 1, 2, and 3, for example, are heavy-duty oils showing apparent alkaline, neutral, and acidic characteristics, respectively, and are all superior in the engine to oil No. 4 which is a neutral, regular-grade oil.

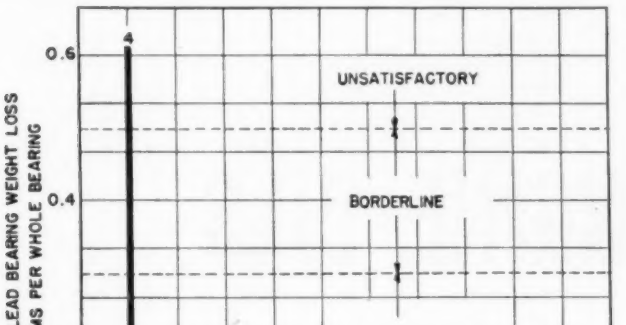
Sligh Oxidation Number versus Engine Performance – Although the Sligh oxidation test is less frequently employed than was formerly the case, there are still some lubricating oil purchase specifications which include a Sligh oxidation number requirement.

Briefly, the Sligh test is a static type of oxidation test in which a 10-g sample of the oil is subjected to oxidation in the presence of pure oxygen for a period of 2½ hr at 200 C (392 F). At the end of the test, the naphtha insoluble content of the oil is determined using precipitation naphtha (ASTM D91-40), and the Sligh oxidation number calculated as the number of milligrams of naphtha insoluble precipitate obtained per 10 g of the oil. The smaller the Sligh oxidation number, the greater the oxidation stability of the oil is *supposed* to be.

The inability of the Sligh test to predict the engine performance quality of oils is illustrated by the data shown in Fig. 2. These data show the results of 36-hr Chevrolet engine tests (same test conditions as related for Fig. 1) on



■ Fig. 1 – Variation of bearing weight loss with acid number – 36-hr Chevrolet engine test – 200 F jacket temperature, 280 F oil temperature, 30 bhp at 3150 rpm



■ Fig. 2 – Variation of bearing weight loss with Sligh oxidation number – 36-hr Chevrolet engine test – 200 F jacket temperature, 280 F oil temperature, 30 bhp at 3150 rpm

a group of oils versus the Sligh oxidation numbers of the same oils. The Chevrolet engine test evaluates the oxidation stability of an oil and its alloy-bearing corroding tendencies by direct observation of the oil's behavior in the engine. The Sligh test subjects the oil to oxidation in laboratory apparatus (glass) under conditions far removed from those in an engine. It is not surprising, therefore, that the Sligh test not only fails to rate oils in terms of their engine performance quality, but may, if used for such purpose, give entirely misleading information. This is particularly so when dealing with straight mineral versus additive-treated oils.

Conradson Carbon Residue versus Engine Performance – As given in the scope of the Conradson carbon residue test (ASTM D189-41), "This method of test is a means of determining the amount of carbon residue left on evaporating an oil under specified conditions, and is intended to throw some light on the relative carbon-forming propensity of an oil. The results of the test must be considered in connection with other tests and the use for which the oil is intended."

In the case of lubricating oils, the test is conducted on a 10-g sample of the oil, and the amount of residue remaining at the end of the test, expressed as percentage of the original sample, is the carbon residue value. The carbon residue value of an oil may be increased when addition agents are incorporated in the oil to improve its engine performance quality. For example, metal-containing additives, which are commonly used in heavy-duty oils, are not completely burned or volatilized in the carbon residue test; the unburned portion remains as a part of the carbon residue.

With respect to engine performance, not only the amount but the nature of the carbon formed by the oil is important. For example, Oil A may show a higher carbon residue value than Oil B. The carbon formed by Oil A may, however, be soft and relatively nonabrasive to such engine parts as aluminum pistons, while that from Oil B, even if smaller in amount, may be quite hard and abrasive to aluminum pistons.

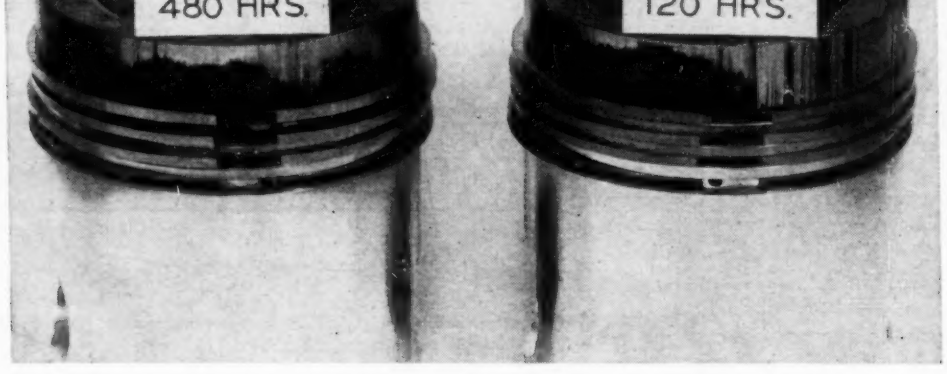
That the carbon residue value may give misleading information if used to predict engine performance quality of an oil is indicated by the piston photographs shown in Fig. 3. These pistons are from single-cylinder Caterpillar (1-A) diesel-engine tests (480 hr, oil changes at 120-hr intervals) on two heavy-duty type of oils of the same carbon residue value. Oil No. 1 shows little to no scuffing, scoring, or line cutting of the aluminum piston, as contrasted to Oil No. 2 which shows severe scoring and line cutting.

Ash Content versus Engine Performance – The incombustible residue which remains on ashing (burning) a sample of new, clean oil, that is, the ash value or ash content of the oil, results from the presence of metal-containing constituents in the oil. These may include addition agents of the following kinds which are added to improve the engine performance quality of the oil:

1. Metal soaps or other metal-organic compounds added to impart increased detergent-dispersant properties to the oil, that is, to promote engine cleanliness.

2. Metal-containing oxidation and/or bearing corrosion inhibitors.

Properly chosen additives of the above types improve the



■ Fig. 3 – Pistons from single-cylinder Caterpillar diesel-engine tests (480 hr, oil changes at 120-hr intervals) on two heavy-duty type of oils of same carbon residue value – oil No. 1 shows little to no scuffing, scoring, or line cutting of aluminum piston, as contrasted to oil No. 2, which shows severe scoring and line cutting

engine performance quality of the oil, although raising the ash content.

The assumption that any increase in the ash content above that for straight mineral oil is undesirable in terms of probable engine performance characteristics is invalid. Neither ash content nor ash analysis figures on the new oil are able to predict engine performance quality. If used for such purpose, the resulting predictions are quite apt to be false and misleading.

For example, it is not possible from a knowledge of the ash content of an oil to predict how the oil is likely to perform in the engine with respect to, say, alloy-bearing corroding tendencies. This is illustrated by the data given in Fig. 4 showing copper-lead bearing weight losses experienced in 36-hr Chevrolet engine tests (280 F oil temperature) on several oils of varying ash content.

Although only a few of the typical oil inspection tests are discussed here, the same conclusions can be reached regarding any single oil inspection test, or any combination of these individual oil inspection tests; that is, that such tests do not furnish a suitable basis for reliably predicting the engine performance quality of oils. This has been well-established by many hundreds of engine tests carried out by both oil companies and engine manufacturers.

The question may then arise, "Why run conventional oil inspection tests on the new heavy-duty oils?"

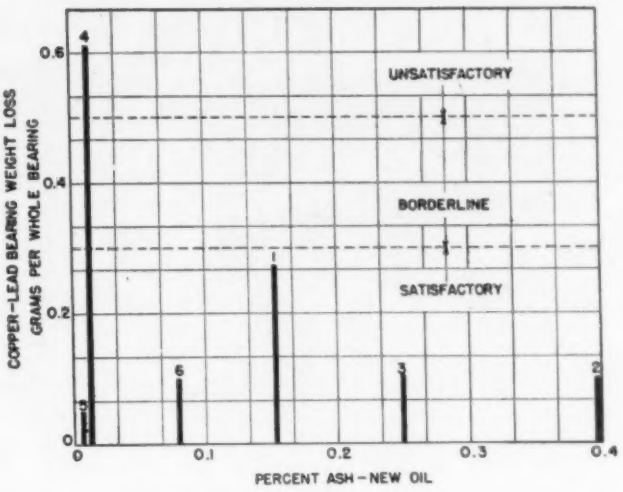
The answer is quite simple: Oil inspection tests are of definite value for purposes of manufacturing control, for checking on the uniformity of successive production lots of the oil, for specifying certain desired characteristics, such as viscosity, viscosity index, and pour point, and for identification of oils. It is only when attempts are made to use the results of these tests for purposes other than those for which the tests are suitable or intended that difficulties and misunderstandings arise.

III. How Can Engine Performance Characteristics of Oils Be Evaluated Properly?

To date the only method known for accurately evaluating the engine performance characteristics of oils is to determine the behavior of the oils in properly chosen engine tests which emphasize the operating conditions to which the oils may be subjected in the particular service or services involved. Final evaluation rests in the engine performance quality of the oil in service.

Probably the most widely used, small, bench test engine employed for preliminary engine testing of oils is the single-cylinder Lauson engine, either in original or modified form. The engine may be equipped with corrodible alloy bearings (copper-lead or cadmium-silver) and may be operated under conditions selected to simulate heavy-duty gasoline or diesel-engine operation. While the test is preliminary in nature, it is quite valuable for research and development work on oils in selecting the more promising oils for further investigation.

Typical results obtained in Lauson engine tests conducted under conditions of full load (3 hp), full speed (1860 rpm) high jacket temperature (340 F), and moderately high crankcase oil temperature (225 F) are shown in the piston photographs of Fig. 5. Piston No. 1 was



■ Fig. 4 – Variation of bearing weight loss with per cent ash – 36-hr Chevrolet engine test – 200 F jacket temperature, 280 F oil temperature, 30 bhp at 3150 rpm

operated for 144 hr on a heavy-duty oil of Specification 2-104B type, SAE 30 grade, while piston No. 2 was operated for 24 hr on an SAE 30 straight mineral oil. The superiority of the heavy-duty oil *under these operating conditions* is readily apparent.

The successful evaluation of the engine performance qualities of heavy-duty oils requires a well-rounded program of engine tests, using both gasoline and diesel engines. Such method of evaluation is followed in both U. S. Army Specification 2-104B and Navy Specification 14-O-13 (INT), as described in two recent and excellent papers on the subject.[1, 2]

Since the reader may not be entirely familiar with these previous publications describing the engine tests employed in evaluating oils to determine their conformance with either of the above specifications, it may be of interest at this point to review briefly the engine test procedures stipulated by U. S. Army Specification 2-104B for the qualification of oils. These procedures are fully described in Coordinating Research Council publications under the designations CRC L-1 to L-5, inclusive.

CRC Designation L-1 – In this test, known commonly as the Caterpillar IA test, the oil is run for 480 hr in a 5¼-in. bore x 8-in. stroke, single-cylinder diesel test engine operated as follows:

Fuel – straight-run diesel fuel of specified characteristics
Speed – 1000 rpm
Oil to Bearing Temperature – 145-150 F
Jacket Outlet Coolant Temperature – 175-180 F
Fuel Input Rate – 2950 Btu per min
Oil Change Period – at 120-hr intervals during the test

The test evaluates chiefly the detergent characteristics of the oil, that is, the ability of the oil to provide clean engine operation, as judged by inspection of the engine parts at the end of test. Among the requirements which the oil must meet to be considered satisfactory in this test are:

No stuck, tight, or scratched rings
No scratches on cylinder liner or piston
Oil rings free of deposits

[1] See SAE Transactions, Vol. 53, April, 1945, pp. 193-200: "Development of Heavy-Duty Oils for Military Vehicles," by W. B. Bassett.
[2] See SAE Transactions, Vol. 53, May, 1945, pp. 269-278: "Testing Heavy-Duty Oils for Navy," by A. D. Brabbs.

■ Fig. 5 – Pistons from Lauson engine tests conducted under conditions of full load (3 hp), full speed (1860 rpm), high jacket temperature (340 F), and moderately high crankcase oil temperature (225 F) – piston No. 1 operated for 144 hr on heavy-duty oil of Specification 2-104B type, SAE 30 grade; piston No. 2 operated for 24 hr on SAE 30 straight mineral oil

Minimum deposit formation in ring grooves, on piston skirt and on the underside of piston.

Minimum piston crown scuffing

Compression rings shall not show excessive feathering or sharpness

Cylinder liner wear shall not exceed 0.001 in. as measured in the prescribed location

Photographs of pistons from CRC Designation L-1 tests on typical 2-104B type, heavy-duty oils are shown in Fig. 6.

CRC Designation L-2 – This test measures the "break-in characteristics and load-carrying ability" of the oil, and is essentially equivalent to Caterpillar test No. 2 (scratch test). The test engine employed is the same as that for the L-1 test except for the precombustion chamber and piston assembly. In this test these parts are designed to force the combustion flame to impinge directly on the antithrust side of the cylinder liner. Test conditions include the following:

Length of test – 3 hr and 20 min

Engine Load – No load for first 10 min; light load for second 10 min; relatively high load for remaining 180 min

Jacket-Outlet Coolant Temperature – 140 F for SAE 10 grade oil; 175 F for SAE 30 grade oil

Oil Temperature to Bearings – 140 F

New liner and rings for each test

Following completion of the first test on a lubricant, the engine is cleaned, a new cylinder liner and new rings are installed, and a repeat test is made. If any scratching of either the liner or rings occurs in either of these first two tests, a third test is then made. An oil is considered to fail the test if scratching of the liner or rings occurs in two of the three tests.

CRC Designation L-3 – This is a test for determining "the stability of an oil and its tendency to corrode copper-lead bearings," and is quite familiarly known as the Caterpillar hot box test (Caterpillar test No. 3A).

The test is run for 120 hr without oil change in a 4-cyl 4¼ x 5½ in. diesel engine operated as follows:

Load – 37 bhp
Speed – 1400 rpm
Oil to Bearing Temperature – 212 F
Intake-Air Temperature – 140 F
Jacket-Outlet Coolant Temperature – 200 F

At least two new cylinder liners, two new piston assemblies, and two new copper-lead bearings are installed for each test.

At the end of test, the power section is examined for ring-sticking, wear, and deposits, and copper-lead bearing weight loss is determined. To be acceptable in this test, the oil must allow no ring sticking and must not permit or cause excessive wear, bearing corrosion, or deposit formation.

CRC Designation L-4 – This is the familiar 36-hr Chevrolet engine test used to determine the oxidation and bearing-corrosion characteristics of oils intended for use under heavy-duty service conditions.

The test is run without oil change, under the following test conditions:

Speed – 3150 rpm

Load – 30 bhp

Jacket-Outlet Coolant Temperature – 200 F

Oil Temperature – 265 F for the SAE 10 grade; 280 F for the SAE 30 and 50 grades

Air-Fuel Ratio – 14.5 to 1

Fuel – 72 octane minimum; 2.5-3.0 ml tetraethyl lead per gal

Prior to test, new piston rings and two new copper-lead bearings are installed.

Criteria employed in rating the performance of the lubricant include:

Extent of deposit formation on engine parts, including both varnish and sludge deposits

Copper-lead bearing weight losses

Condition of the used oil as determined by analysis of samples taken at specified intervals throughout the test

CRC Designation L-5 – This procedure is used "for determining in an engine the oxidation, ring-sticking, detergency, and bearing-corrosion characteristics of engine crankcase oils designed for use under heavy-duty service conditions."

The test is run in a GM Model 71 diesel engine (3 or 4

cylinders) for a period of 500 hr with no oil change. Other specified test conditions are as follows:

Speed – 2000 rpm
Output – 26 bhp per cyl (approx.)
Oil Temperature – 230 F
Intake-Air Temperature – 105 F
Jacket-Outlet Coolant Temperature – 180 F

Prior to test the engine is reconditioned by fitting with at least one new piston, new injector cam followers, new piston rings, and new copper-lead bearing inserts.

"Performance of the test oil is judged by examination of the power section for ring-sticking, deposits, and wear, by noting the corrosion of the bearings, and by inspection of oil samples taken during the test run."

Under Specification 2-104B, CRC Designation L-5 is specified only for the SAE 30 grade oil, and only when the finished engine oil uses an additive not previously used in an engine oil qualified by the Ordnance Department.

IV. When and How Should Heavy-Duty Oils Be Used?

As the name implies, heavy-duty oils were developed to provide satisfactory engine lubrication under heavy-duty service conditions where engine design and operating factors are such as to impose unusually severe demands upon the lubricating oil. Such design and operating factors include engine speed and load, the use of alloy bearings (such as copper-lead and cadmium-silver) and sustained operation at high engine and oil temperatures.

Fortunately, the cooperative work which has been conducted jointly by the engine manufacturers and the oil industry during the period of development of improved engines and lubricants has established quite definitely the lubricating oil requirements necessary for optimum performance in both gasoline and diesel engines in heavy-duty service. Based on this background of cooperative work, which comprised extensive laboratory and field service tests, the best answer to the question, "When should heavy-duty oils be used?" is to be found in the recommendations of the engine manufacturer and the oil supplier for the particular engine and service involved.

Generally speaking, heavy-duty oils of Specification 2-104B type are recommended particularly for applications which require a lubricant possessing the following combination of properties:

1. A high degree of detergency-dispersancy – to prevent ring-sticking and piston deposits.

2. High oxidation resistance – to prevent lacquer and varnish formation.

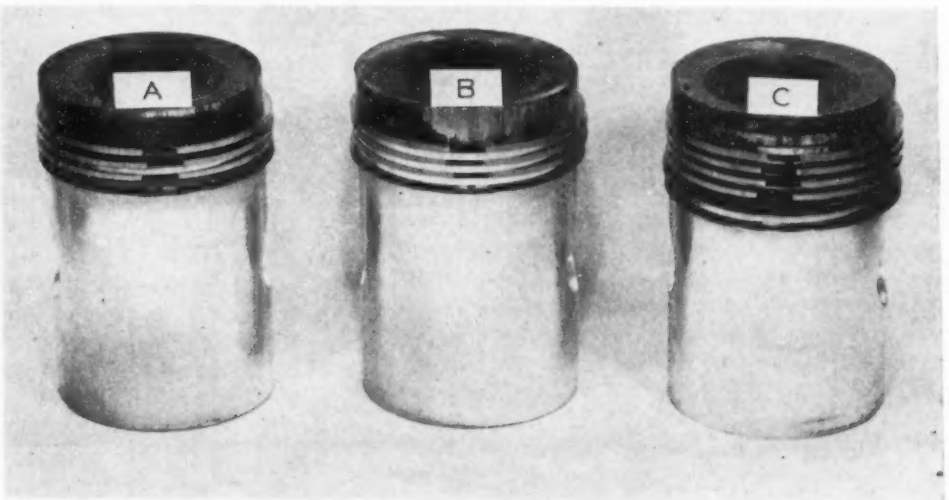
3. Noncorrosiveness to alloy bearings.

While heavy-duty oils of the type discussed herein represent a large step forward in the development of improved lubricants for heavy-duty service, it does *not* follow that such lubricants are now required for satisfactory operation of all engines in all types of civilian service. All such service is not extreme heavy-duty service, and all oils giving satisfactory service performance are not heavy-duty oils. There is still a large number of engines in service which are being satisfactorily lubricated with good quality, straight mineral oils, or with oils stabilized against oxidation and bearing corrosion but containing little or no added detergency.

As work progresses and further information is gained concerning the lubricating oil requirements for various engines and services, it may be that oils lying between straight mineral oils, on the one hand, and 2-104B type heavy-duty oils on the other, will be found especially advantageous for intermediate or semiheavy-duty service. Such oils may embody appropriate adjustment of the relative detergency versus oxidation stability characteristics.

■ Need For Proper Maintenance

With regard to the question, "How should heavy-duty oils be used?" there appear to be several points worthy of emphasis. First, the use of heavy-duty oils does *not* constitute a cure-all for all engine troubles. Troubles arising from faulty installation of bearings, leaking cylinder-head gaskets, excessively long periods between oil drains, improper fuel or fuel combustion, long periods of operation under heavy overload, and the entrance and accumulation of harmful quantities of abrasive contaminants in the lubricating oil (through inadequate filtration of oil and intake-air, and too long a period between oil drains) are not attributable to the lubricating oil, and cannot, of course, be cured merely by changing from regular to heavy-duty type of oil. As with any oil, the optimum performance



■ Fig. 6 – Pistons from CRC Designation L-1 tests on typical 2-104B type, heavy-duty oils

with heavy-duty oils can be obtained *only* in combination with proper maintenance procedures.

■ Oil Change Schedule

Next, what oil change schedule is most desirable in the case of heavy-duty oils? The answer to this question depends, of course, upon a number of factors, including: type and severity of operation, engine design and mechanical condition of the engine, the degree of crankcase oil filtration, and the kind of inspection and maintenance practices which are followed. Since these factors are variable, no single answer in terms of miles or hours of operation between drains can be given to cover all cases. However, from a background of research information and experience, the engine manufacturer and oil supplier are in a position to recommend the proper oil drain period for the particular engine, lubricating oil, and service involved. Both the engine manufacturer and oil supplier are equally interested in attaining the same goal, that is, the best performance, and their recommendations as to oil change practice are based on their best available information as to how this goal of optimum engine performance can be reached. Because the additives included in the oil are used up in service, attempts to extend the oil drain period appreciably beyond that recommended may result in a false economy.

Related to the question of oil drain period is the matter of what procedure should be followed in changing to the use of a heavy-duty oil in engines which are in service and which may be in quite dirty condition. This item has been discussed in several recent papers.[3, 4, 5] If the engine is not down for complete disassembly and overhaul – which would permit a thorough cleaning to be made – the best practice is to drain the old oil from the engine while hot, and then drop the crankpan; where consistent with engine design, the valve, rocker arm, and push-rod cover plates should also be removed. Dirt and sludge deposits in the areas thus exposed should then be removed as completely as possible. Where neither of these procedures can be followed, the old oil should be drained while hot, immediately following a period of engine operation, and the engine should then be cleaned as thoroughly as possible by a suitable flushing procedure. Following introduction of the heavy-duty oil, the oil pressure should be observed closely during the first few oil drain periods in order to detect immediately any significant drop in pressure which might occur through possible plugging of the oil pump screen or oil passageways. This may occur when dirt or sludge remaining in the engine from previous service is of the type susceptible to loosening up by the detergent-dispersant action of the heavy-duty oil. New oil filter cartridges should be installed when the change to heavy-duty oil is made, and subsequently the cartridge should be renewed at least each time the oil is changed. The first few oil drains should be made at frequent intervals (500 miles or less, depending upon the type of service in which the engine is used and the length of time since the last overhaul). The oil drain period should then be extended moderately for the next one or two drains, proceeding finally to the regular oil drain period to be established for the service in question.

[3] See *Petroleum Refiner*, Vol. 23, December, 1944, pp. 124-128: "Detergency or Dispersancy in Heavy-Duty Engine Oils," by C. W. Georgi.

[4] See SAE Transactions, Vol. 53, February, 1945, pp. 79-82: "Cold-Engine Sludge and Its Control," by B. E. Sibley.

[5] See *Petroleum Engineer*, Vol. 15, May, 1944, pp. 204, 206-208: "Heavy-Duty Engine Oils," by R. M. Welker.

■ Sludge Formation

Concerning the question of cold-engine or winter sludge formation, it is incorrect to assume that such troubles can be magically cured by changing to a heavy-duty oil and neglecting to apply other corrective measures bearing more directly on the source of the difficulty. Figuratively, some medicines may ease the pain but fail to cure the disease. Similarly, heavy-duty oils may alleviate cold-engine sludge formation, but be unable to cure this engine disease without necessary assistance in the way of proper engine and oil temperatures, sufficiently frequent oil drains, and adequate crankcase ventilation. It is still true that oil quality is one of the least important factors bearing on the control of winter sludge formation. Winter sludge formation may occur with any oil when operating and maintenance conditions are such as to result in contamination of the crankcase oil with excessive amounts of partially burned fuel residues, fuel soot, and water (from combustion of the fuel, leakage past the gaskets, crankcase breathing, and sweating). When these contaminants, together with other contaminants which may be present, for example, dirt and lead compounds from the combustion of leaded fuel, are churned in the oil during operation under cold-engine conditions, a thick, viscous, mayonnaise or emulsion type of sludge is formed, that is, cold-engine or winter sludge.

As concluded by B. E. Sibley in his recent paper,[4] "The use of the heavy-duty oils is considered as supplementary only to the items just mentioned in controlling sludging difficulties to a satisfactory minimum (good supervision in maintaining healthy or normal engine temperatures, the use of a good quality oil, in combination with effective filters, air cleaners properly serviced, observance of a sane crankcase drain period, and the practice of draining the crankcase hot)."

■ Filters

Finally, any good maintenance procedure designed to obtain optimum performance of engines and oils will include the use of proper oil filters and filter cartridge change periods. In the case of heavy-duty oils particularly, the oil supplier should be consulted as to the type of filter he recommends for the oil in question. Preferably, the filter cartridge should be renewed at least as often as the oil is changed.

■ Conclusion

In conclusion, the present paper has attempted to touch upon some of the high spots concerned with the nature, evaluation, and use of heavy-duty oils. In so doing, much of the material presented covers information already available in the literature on the subject. However, if such repetition succeeds in advancing even a little closer to the goal of optimum service performance, the paper will indeed have served its purpose.

Cruising Economy by Use of Water Injection

by D. C. EATON
Wright Aero, Ltd.

A STUDY of possible economic advantages to the aircraft operator of utilizing the internal cooling potentialities of water to assist aircraft-engine cooling and increase detonation limited powers is presented here.

Cost analysis is made of three methods of increasing airplane performance by water injection, namely:

1. By increasing cruising bmep's while maintaining engine cooling requirements at the value for conventional cruising bmep's.

2. By decreasing cruise power cooling requirements with the addition of water.

3. By increasing cruise powers up to meto at a constant F/A ratio while maintaining low cooling requirements and avoiding detonation with the use of water.

It is concluded by the author that if the water to be used for cooling does not displace an equivalent weight of payload, a maximum saving of 7% at sea level and 18% at 25,000 ft can be realized in direct operating cost with water. At the same operating cost a performance increase of 3% at sea level and 12% at 25,000 ft can be realized. However, if the water carried results in an equivalent loss in payload, the operating cost may be doubled if water injection is used.

THE AUTHOR: D. C. EATON, who is at the West Coast office of Wright Aeronautical Corp., has been primarily concerned with the Consolidated B-32 program. Mr. Eaton has been with Wright since 1940, becoming field engineer in 1942, spending time at Grumman, Republic and General Motors until he was assigned the same year to work with the British Air Ministry and U. S. Air Technical Section in England. He is a graduate (1940) of Harvard College. He has been with Wright Aero, Ltd., since his return from the European Theater.

THE purpose of this paper is to report the results of an investigation into the benefits to an aircraft-engine operator of utilizing the internal cooling properties of water. Although undertaken entirely independently, it will be apparent that this study is supplementary to the investigations reported by Rowe and Ladd.[1] In this paper the authors established the basic characteristics of aircraft-engine operation with antidetonant injection. Although this paper was primarily concerned with operation at higher than take-off powers, a part was devoted to the advantages to be gained with use of water at powers below meto. It is the author's intention to construct on the basic premises established by these and other investigators an analysis of direct flying costs in dollars and cents when water is used to replace fuel as an internal coolant.

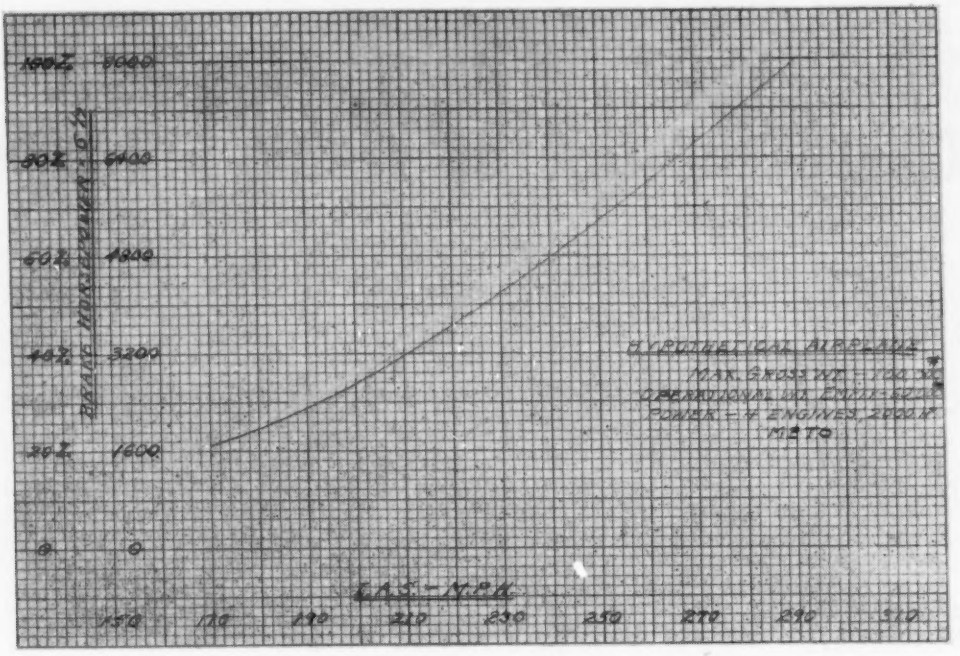
Sufficient evidence is available in the form of reports and papers presented before this and other technical organizations, so that it is considered unnecessary to substantiate here by theory or test experience the basic assumptions utilized in this study. Instead the premises will be established in the form of assumptions which can be demonstrated by reference to other work in this field.

In the first place, since it is desired to evaluate the benefits of replacing with water all excess fuel not required for efficient combustion, it is necessary to establish a F/A ratio which will provide good combustion at all powers for a typical engine, yet not furnish the excess fuel normally required for internal cooling. In selecting this ratio, examination of combustion data from a group of representative engines at all powers was made. Since the minimum operable F/A ratio for any engine is generally determined by distribution and stability characteristics rather than by detonation or cooling, a considerable variation in minimum F/A ratio is found. A second consideration is the necessity for choosing a ratio reasonably near the value for best power in order to obtain the optimum horsepower output per inch of map. With the addition of water to the combustion cycle of an engine the specific air consumption is slightly increased, while the fuel flow for best power remains constant; therefore, the best power F/A ratio is reduced. In view of these factors a value of 0.070 F/A ratio was chosen for this study. This ratio is slightly above the minimum value and at the same time within less than 10% of the best power fuel-air ratio of all the engines investigated.

Secondly, it is assumed that water is as effective as 100-octane fuel as an internal coolant, although actual test experience has indicated that this is not always true.

[This paper was presented at a meeting of the Southern California Section of the SAE Los Angeles, Nov. 1, 1945.]

[1] See SAE Transactions, Vol. 54, January, 1946, pp. 36-48: "Water Injection for Aircraft Engines," by M. R. Rowe and G. T. Ladd.



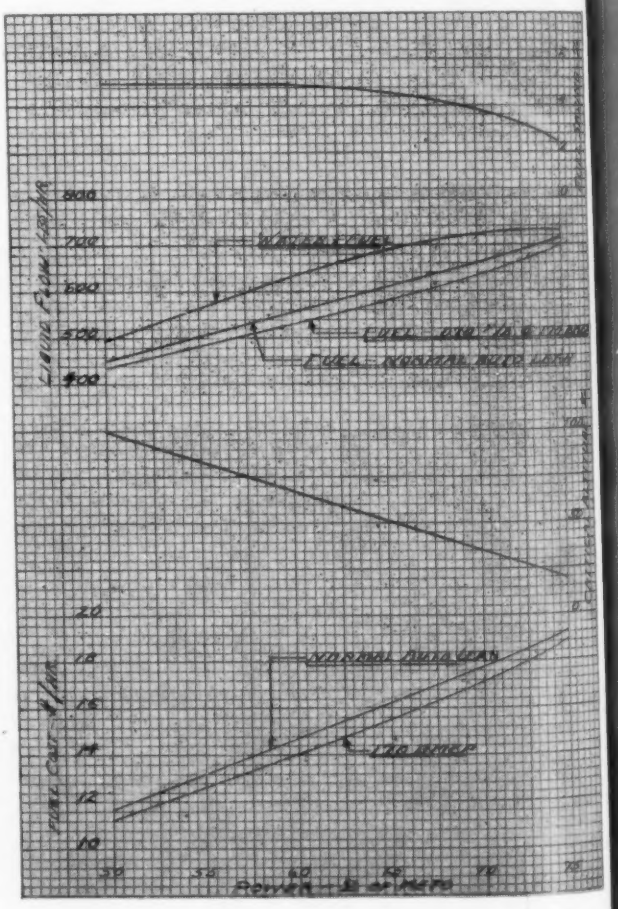
■ Fig. 1 – Power required curve

In addition to these basic assumptions it is necessary to hypothesize a typical airplane and powerplant. The characteristics of this airplane are postulated as follows:

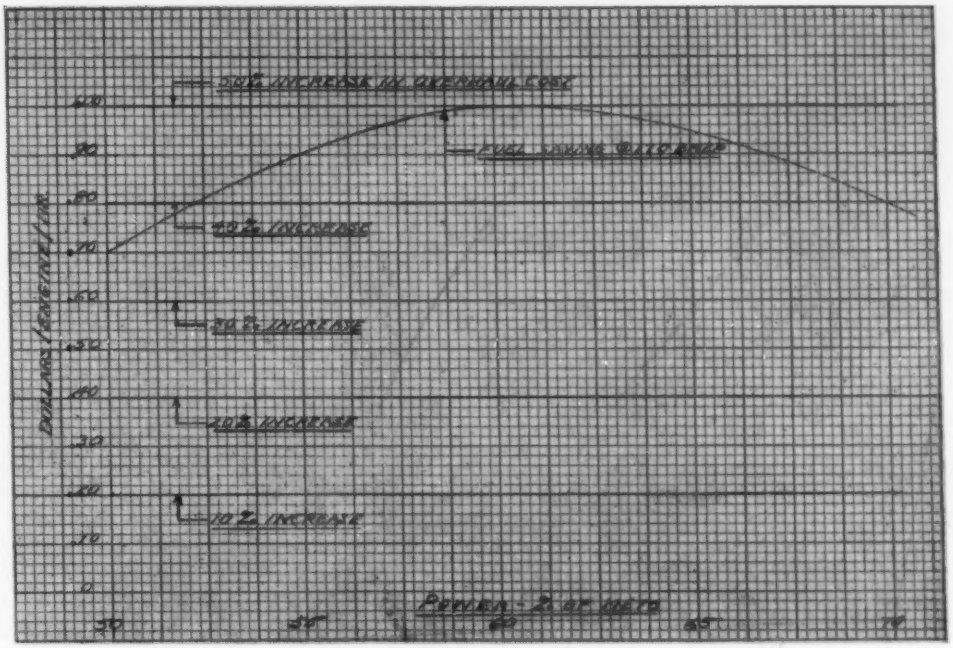
4 engines, 2000 hp, meto
100,000 lb maximum gross weight
Performance as illustrated in Fig. 1

There are a great many ways in which the cooling potential of water may be realized. It may be used to increase the detonation limited powers with low octane fuel. Its use might be anticipated in establishing engine design criteria in order to realize the benefits of compression ratios, spark settings, combustion-chamber design, and other factors affecting the combustion cycle which are now limited by the detonation characteristics of current fuels. Water can also be used to allow an increase of the allowable imep at cruise powers without destructive detonation or excessive cooling penalty. The cooling properties of water can be used to reduce the cooling requirements of the engine and so increase the net thrust horsepower available to the airplane. Finally, water can make possible the use of cruising powers up to the engine's maximum allowable continuous power without paying the price differential for the fuel normally required for internal cooling.

Of these potentialities the use of water to increase the power output of engines designed for low octane fuel has been quite thoroughly investigated previously. The operation of engines designed for 100-octane fuel but operated on lower octane fuels with water will show economic benefits equal only to the difference in cost of the fuel used. In other words, since no gain in airplane performance is obtained, the saving in using 91- instead of 100-octane fuel would be 3¢ per gal. However, a serious handicap is incurred in both of the above cases because the engine must have water to operate at the powers on which airplane performance and operating schedules are based. Therefore, the liquid system hazard is doubled since failure of either fuel or water system will dangerously compromise the performance of the airplane. On the other hand, if the engine is allowed to operate on the grade fuel for which it was designed, and the benefit realized by a reduction in F/A ratio with water, then if the



■ Fig. 2 – Comparative data – 170 bmep versus normal auto lean



■ Fig. 3 – Effect on profits of increase in overhaul cost per flying hour.

water system fails the penalty is apparent in increased operating cost rather than in reduced performance.

The second of these five possibilities cannot be evaluated in this study because of the many unknown factors involved and the extremely complicated relationship of each design factor to the other. Therefore, this paper will be confined to a discussion of the last three items mentioned.

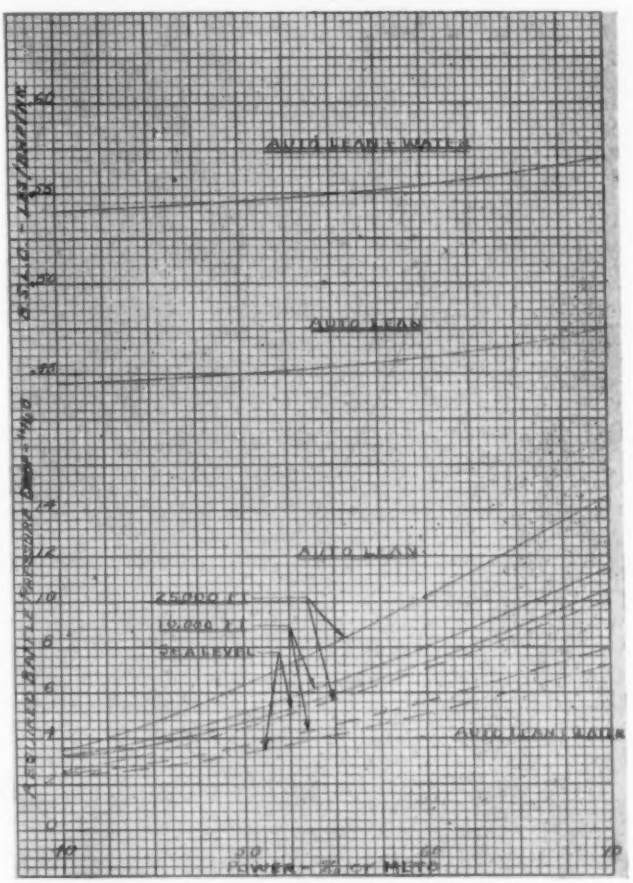
■ Schedule A

Increasing the detonation limited bmep of an engine operated on 100-octane fuel by the use of water as detonation inhibitor and coolant:

The increase in economy in operating to a cruising power schedule based on high bmep rather than the conventional 140 or 145 bmep is basically derived from the reduction in friction horsepower obtained by decreasing the engine speed for constant power. In order to capitalize on this engine characteristic it is necessary to operate at high bmep at the F/A normally established for 140 bmep. Since 140 bmep is conventionally limiting because of both detonation and engine cooling it is necessary to add sufficient water to suppress detonation and maintain engine cooling requirements at the lower value.

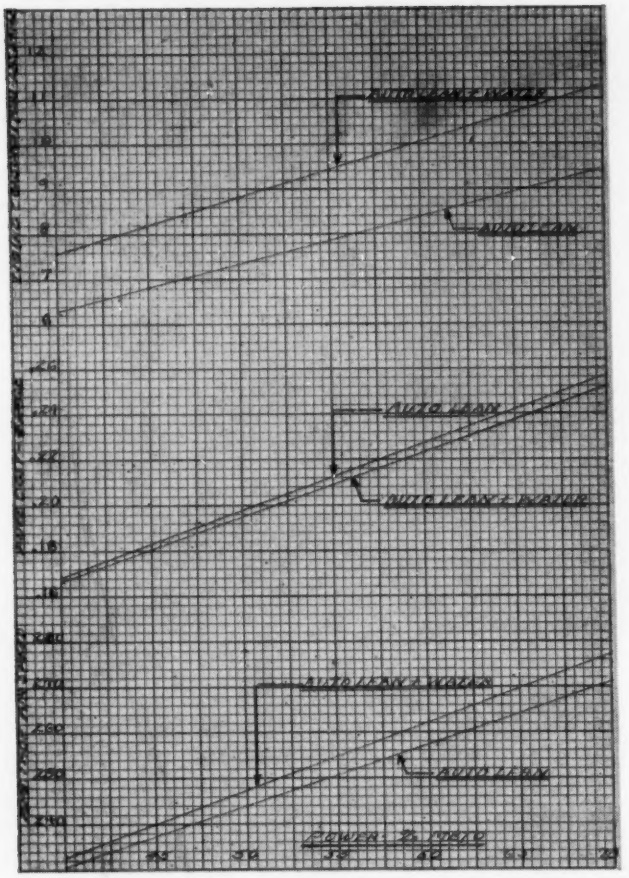
Engelman and White[2] have evaluated the amount of water necessary to accomplish this increase for a Cyclone 2600-8 engine. It was found necessary to increase the liquid consumption about 15%, while maintaining a constant fuel-air ratio, to obtain detonation-free operation with no increase in head temperature when the bmep was increased from 140 to 170.

Figure 2 illustrates the amount of fuel saved per hour in dollars and in pounds when this power schedule is applied to our hypothetical powerplant. It will be noted that the percentage savings are constant up to 60% power, and decrease above that value. This is explained by the fact that at higher powers the normal power schedule allows operation on a propeller load curve. The normally



■ Fig. 4 – Comparative data – auto lean with water versus normal auto lean – CAA summer day

[2] See NACA Research Bulletin No. E4H12, August, 1944, "Use of Water Injection to Decrease Gasoline Consumption in an Aircraft Engine Cruising at High Power," by H. W. Engelman and H. J. White.



■ Fig. 5 – Comparative data – normal auto lean versus auto lean plus water – 10,000 ft – CAA summer day

allowable bmep values become greater than 140 at speeds above the rpm at which a constant 140 bmep line intersects the propeller load curve. Therefore, above 60% power, the difference and the cost benefit between our proposed 170 bmep and the conventional schedule becomes less as the power increases.

There are two additional factors, one known and one unknown, which make the value of this small fuel saving extremely questionable. The first is the very appreciable loss in critical altitude which is incurred when the engine speed is reduced for a specific power. This loss is greater as the specific power becomes less, and is 100% at 50% power (Fig. 2). What this amounts to is that for 50% power at 170 bmep our critical altitude is sea level instead of 9000 to 10,000 ft.

The second factor is one which cannot be evaluated at this time. This is the reduction in service life caused by continuous operation at high bmep. It is apparent that there must be some penalty resulting from the added strain on engine parts. Although it cannot be estimated just what this penalty would be, we can easily show what its effect will be in dollars and cents.

Fig. 3 illustrates the effect on our profits of the increase in overhaul cost per flying hour. The curve shows that if the increase in overhaul cost is 50% the net profit will be zero if we fly at 60% power and 170 bmep. This, of course, does not take into account the increase in expense of ground facilities resulting from the higher frequency of engine changes.

■ Schedule B

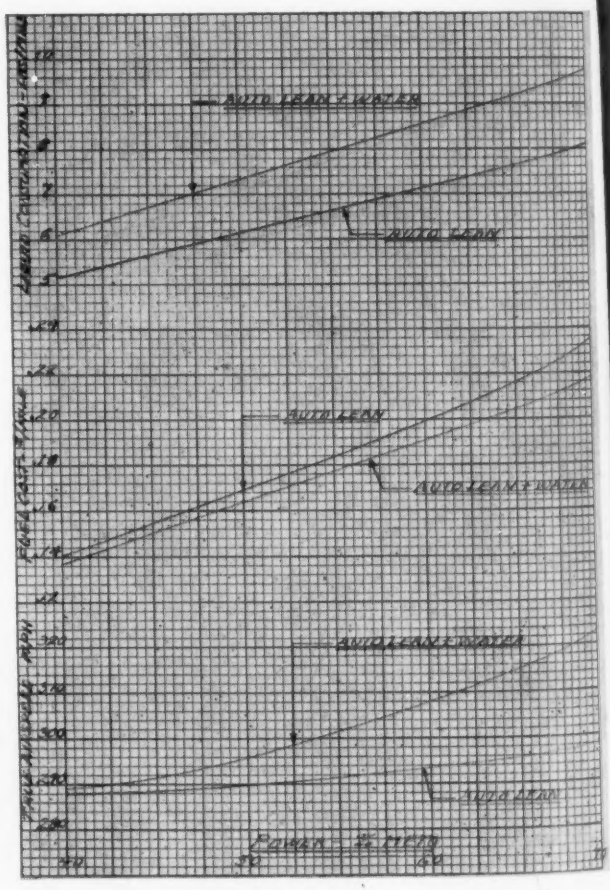
Decreasing the engine cooling requirements at cruise powers by adding water to auto lean mixtures:

In this case the economic benefit is obtained by an increase in air speed by virtue of the added thrust horsepower available. Therefore, a plot of engine data alone will not be of any final value. However, by calculating the actual decrease in baffle pressure drop required to maintain a constant head temperature and the weight of water necessary to accomplish this decrease we have data which can be translated into dollars and cents with the aid of our hypothetical airplane.

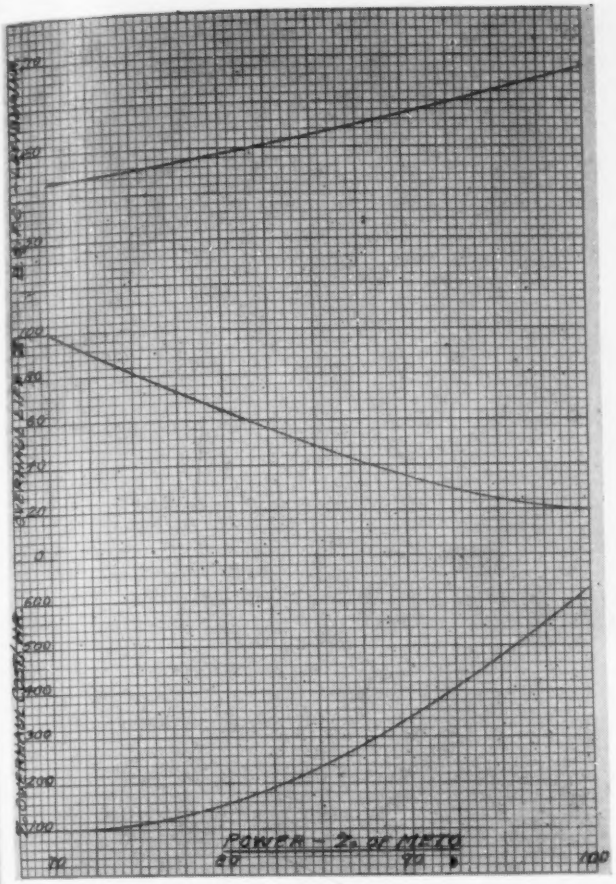
In order to simplify the problem, the calculations were based on the addition of sufficient water to make the cooling requirement equal to the normal auto rich requirements. In this way both the baffle pressure drop and the water flow are established, since by our second assumption water is equivalent to fuel as an internal coolant.

Fig. 4 illustrates the basic liquid consumption which will be required to duplicate auto rich conditions. Fuel (at 0.070 F/A ratio) will form approximately 80% of the total flow, with water making up the remainder.

Fig. 4 also illustrates the baffle pressure drop required to maintain a 450 F maximum head temperature in auto



■ Fig. 6 – Comparative data – normal auto lean versus auto lean plus water – 25,000 ft – CAA summer day



■ Fig. 7 – Typical engine characteristics – 70 to 100% meto with auto rich mixtures

lean and auto rich from sea level to 25,000 ft on a CAA summer day. An indication of the advantage in decreased cooling drag which may be gained is shown by the fact that the auto rich baffle pressure drop at 25,000 ft is a little less than that required at sea level in auto lean.

Applying these data to our airplane, it is possible to calculate the weight of fuel and water, the ground speed, and cost of fuel per mile for operation in auto lean with and without water. The results of these calculations at 10,000 ft are shown in Fig. 5. At 70% power the increase in speed and decrease in cost are about 3% and 7½%, respectively.

Fig. 6 shows the same data for 25,000 ft. These data are based on the assumption that a turbo is available to maintain low blower horsepower to this altitude. However, no cooling penalty has been assumed for the higher exhaust back pressures resulting from the use of a turbo. Calculations for a two-speed engine show that the penalty is greater, but for the purposes of this paper the introduction of high blower characteristics unnecessarily complicates the presentation of comparable data.

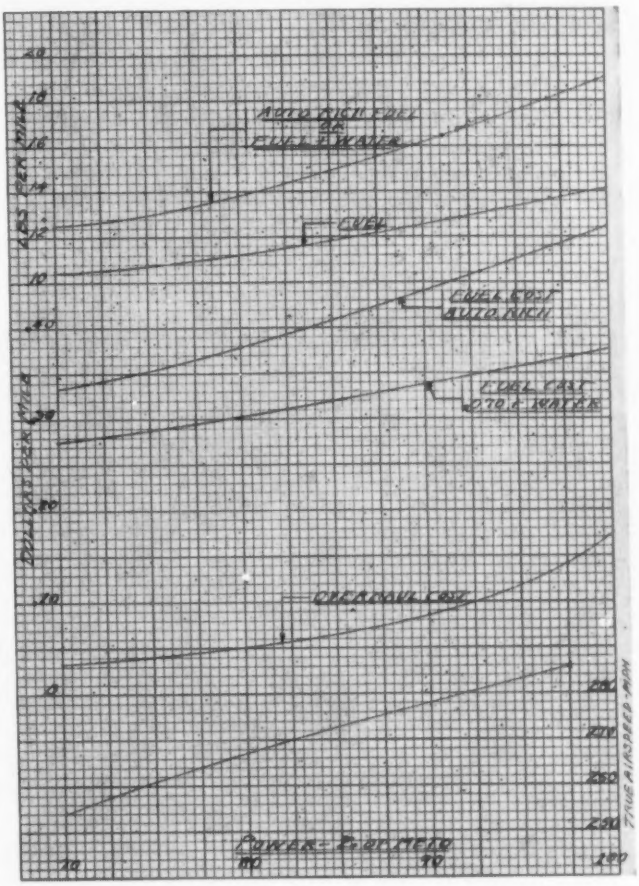
The increase in speed and consequent decrease in cost are a little over 12% and 18%, respectively, at 70% power for this altitude. This, at first examination, seems excessive. For some airplanes it may be. For others it may be conservative. It is difficult to generalize the characteristics of airplanes as regards the effect of cooling drag. Theoretical calculations are at best controversial and of doubtful accuracy. It is well known that at high altitudes it is possible to obtain a *decrease* in air speed when increasing power in auto lean. This, of course, is caused by the fact that under some conditions the cooling drag increases faster than the thrust horsepower. Under the best conditions the increase in speed is far less than the conventional cube relationshbip would indicate. The results presented here are based on actual test data obtained on a conventional 4-engine airplane.

■ Schedule C

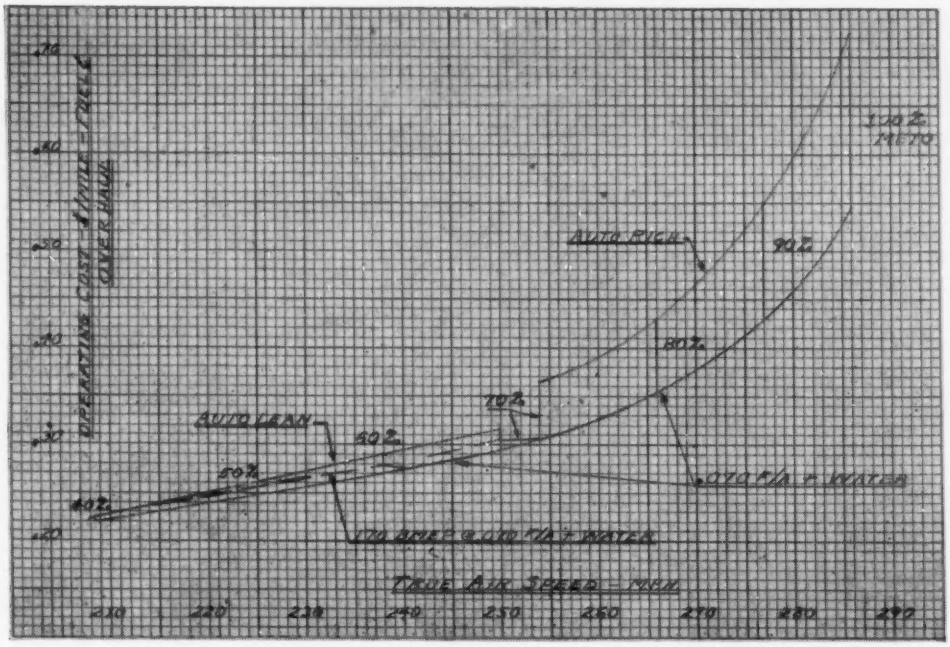
Increasing cruising horsepower up to meto power at a constant F/A ratio of 0.070:

Cruising operation at powers above 70% of meto would be a radical departure for commercial airlines although the Air Forces have thought little of using 80, 90, or 100% rated power for continuous cruising. However, in wartime the operating cost in dollars has no bearing on the choice of power schedules. It remains to be seen whether it is possible for the cost-minded operator to make economical use of the higher ranges of powers.

Obviously, another operating cost factor enters into the picture when we consider exceeding 70% power for long periods. Guarantees of overhaul life and cost are presently established by the engine manufacturer on the assumption that the operator does not intend to use more than 70% power for extended cruising. For cruising powers such as are contemplated here an upward revision of overhaul costs



■ Fig. 8 – Comparative cost data – normal auto rich versus 0.070 F/A plus water – sea level



■ Fig. 9 – Comparative costs – sea level

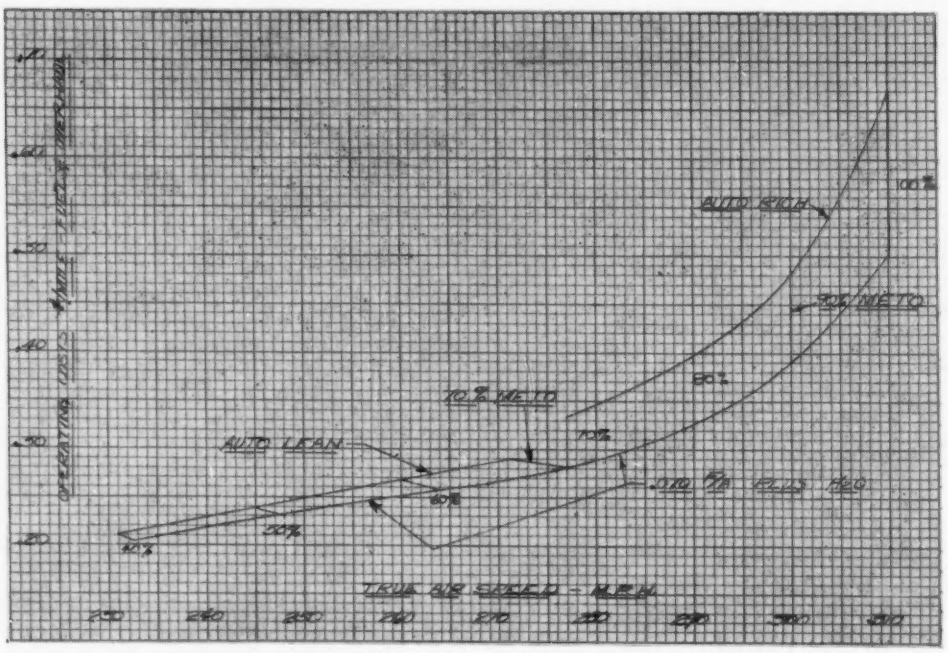
will be necessary. Fortunately, studies have been made of data accumulated over a long period of time, and an estimate of the cost increase plotted against power is available. It should be remembered that this estimate does not apply to any specific engine and is used here only as a general yardstick for computation of our typical operating cost.

In Fig. 7 are shown the characteristics of a typical engine at powers between 70 and 100% of meto. The information pertinent to this discussion is the fuel consumption, overhaul life, and overhaul cost per flying hour (in per cent of the time and cost at 70% power). The overhaul cost rises to 625% at meto power, while the service life decreases to 20% of that at 70% power. This is a very severe penalty which arises both from the reduction in flying time between overhauls and the increase in the number of parts requiring replacement at overhaul.

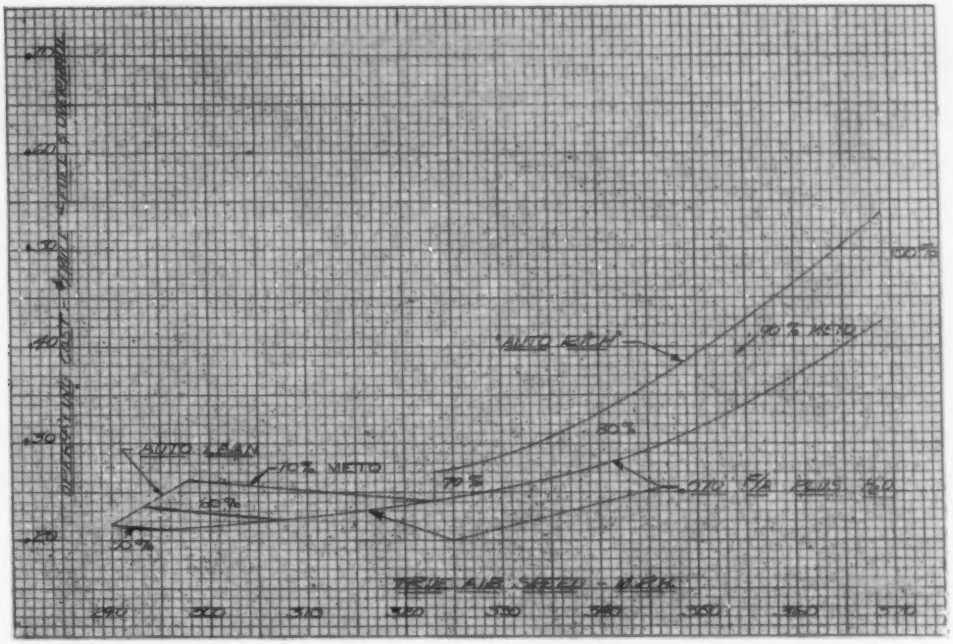
It can be seen at the outset that we have a large increase in cost to overcome before the operation can be an economic success.

Our calculations for fuel and water consumption are based on a constant 0.070 F/A ratio throughout the power range. For determining the speed of the airplane we will again assume, for simplicity, that we will supply enough water to make the cooling requirement equivalent to normal auto rich operation.

Fig. 8 shows the fuel and water consumption, the fuel



■ Fig. 10 – Comparative costs – 10,000 ft



■ Fig. 11 – Comparative costs – 25,000 ft

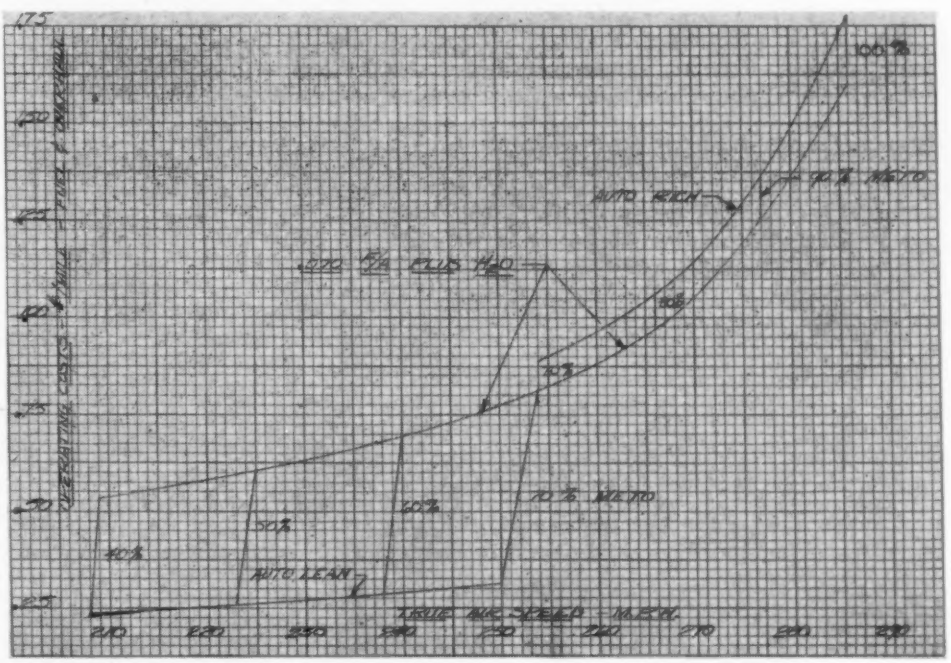
and overhaul cost, and the ground speed for normal auto rich and for operation at 0.070 F/A ratio with water at sea level. It is apparent that for any power above 70% it costs more to operate with water than to operate at 70% auto lean, but that a very substantial saving in fuel cost is realized when compared to normal auto rich power settings.

This completes the detailed discussion of the three methods of water utilization pointed out earlier in this paper. However, before concluding, it remains to summarize the various gains and losses of the proposed operating techniques in a form both more comprehensive and comprehensible. It is not difficult to do this in a relatively simple form.

Fig. 9 illustrates the total direct flying cost per mile at sea level plotted against true air speed. From this curve can be obtained the true air speed and the cost of flying according to any of the five cruising techniques discussed above. Thus, it can be seen that for a cost of 28¢ per mile the operator can fly at 233 mph at 57% power with conventional auto lean mixtures, or at 238 mph at 59% power using 170 bmep, or at 242 mph at 62% meto at 0.070 F/A ratio plus water. If he wishes to adopt high power cruise schedules, he can operate at 257 mph at 73% power for the same cost as he can fly at 70% power in auto lean.

Figs. 10 and 11 represent the same data for 10,000 and

concluded on page 92



■ Fig. 12 – Total operating costs if water displaces payload – loss in payload based on $0.45 per ton-mile – sea level

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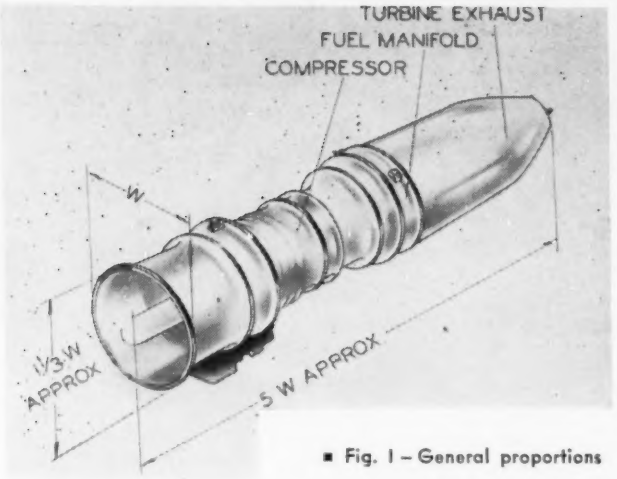

■ Fig. 1 – General proportions

ALL outstandingly successful airplane designs have at least one feature in common. This feature is that the engine delivers its predicted performance or better when coordinated with an installation arrangement which satisfies the specified engine and airplane operating conditions safely and efficiently. Coordination referred to is defined as a "harmonious adjustment or functioning" and it must be attained in the projects now in progress and in designs of the future. Achievement of coordination of the engine and its installation requires a great deal of mutual understanding of the problems of the customer, the airplane manufacturer, and the engine manufacturer, and close liaison among the three. The division point of design responsibility between the engine and its installation provisions will then be wisely chosen to ensure the best complete airplane. The result will certainly be a very gratifying reduction in the amount of time and effort necessary to develop fully operational aircraft. With these basic thoughts in mind, let us review in systematic order the problems of installing a typical, axial-flow type of aviation gas turbine engine.

At this point in a revolutionary engine development program, improvements are being made at a rapid rate, and this review will be a quick glimpse of a constantly changing scene.

The place of the gas turbine type of engine in the industry has already been accurately indicated. This discussion will, therefore, be devoted to the problems associated with very high-speed aircraft where this type of engine is best suited. Since the turbo-jet engine was the first developed, we will deal primarily with its installation and add briefly to this the problems encountered in turbine-propeller engine installations.

■ Basic Arrangement

Before trying to discuss the problems of installing a turbo-jet engine it is necessary that one get a clear picture of the proportions and basic arrangement of the machine whose operating requirements must be satisfied.

[This paper was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Oct. 4, 1945.]

In Fig. 1 is shown an outline of a typical engine and attached accessories. The basic frame is round and usually about five times as long as it is in diameter. This makes the height, including accessories, approximately one-third greater than the width. The accessory section may be rotated to the side or top to obtain a better arrangement, if desired. The accessory envelope is compact and fairs in very nicely with the aft end of a fuselage, or an enclosure of good aerodynamic shape, without protrusions. The small frontal area of axial-flow turbo-jet engines makes their application to aerodynamically clean airplanes very favorable. The small size permits location of the engine, its enclosure, inlet ducting, and exhaust nozzle so as to give critical Mach numbers approaching the critical Mach number of the wing-fuselage combination without the engine, engine enclosure, or ducting. This problem is primarily aerodynamic and the high Mach number investigations of various types now in progress will provide additional data for future designs. As far as engine location is concerned, aerodynamically efficient airplanes can be de-

THE author divides this discussion of the problems likely to be encountered in the installation of the turbo-jet engine to very high-speed aircraft as follows:

1. Coordination of the design of the engine air inlet duct so as to bring the air to the engine inlet with minimum loss in total pressure and maintain even distribution across the face of the engine inlet.
2. Engine suspension.
3. Engine fuel system.
4. Engine lubrication system.
5. Instrumentation necessary to check proper engine operation.
6. Cooling of engine accessories.
7. Turbo-jet engine control system.

He also mentions briefly the problems involved in turbine-propeller engine installations.

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GAS TURBINE INSTALLATION PROBLEMS

by M. C. BENEDICT

Section Engineer in Charge of Installation
Aviation Gas Turbine Division
Westinghouse Electric Corp.

signed with the engine or engines located to give the best overall arrangement of the airplane.

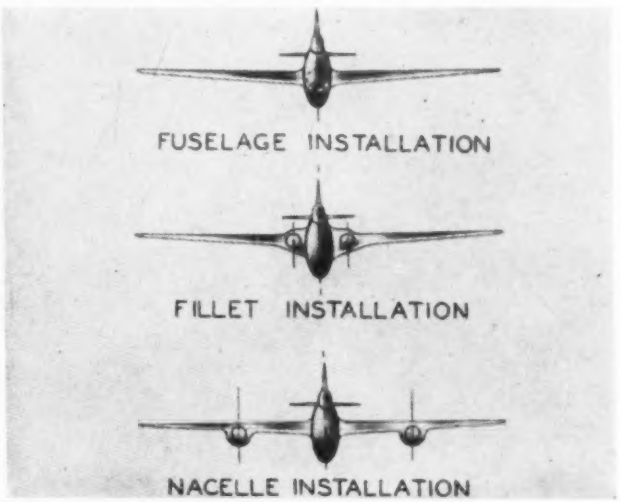
To illustrate this point more clearly, let us look at a few possible arrangements in small, fast aircraft, as shown in Fig. 2. The first is a turbo-jet engine located in the fuselage with the duct inlets in the wings, the fillets, belly scoop, nose, or sides. A second shows the engines located in the fillets with the duct inlets directly in front. The last shows nacelle locations in the wings. Any of these arrangements could be used in an airplane which also has a reciprocating engine. There are other locations, such as in or on the top of the fuselage and in the belly, which might be desirable for certain applications. Engine location in medium- and large-sized aircraft will not be discussed, since the problem is similar and engine location can be chosen to give the best overall configuration.

■ Inlet Air Duct

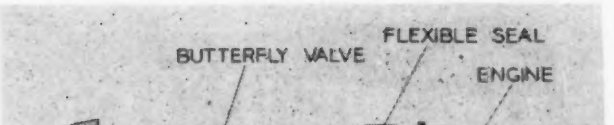
The inlet air duct for an axial-flow aviation gas turbine engine is shown schematically in Fig. 3. The problem of coordinating the design of the engine air inlet duct is one which when properly solved will do most to ensure the ultimate in engine performance. The objective is to bring air to the engine inlet with a minimum loss in total pressure and maintain even distribution across the face of the engine inlet. One of the unusual characteristics of the gas turbine engines is that they handle very large volumes of air. Since the engine inlet is small in area, the air travels at high velocity. The engine sucks air when the airplane is at rest on the ground and at low flight speeds, but as forward velocity is increased, the condition is usually reached at about cruising speed, where the speed of the airplane equals the velocity of air entering the engine. At higher flight speeds, ram pressure is developed and the excess air is deflected around the airplane. The ratio of duct inlet area to engine inlet area selected should provide the best compromise performance to suit the operational requirements of the airplane. The result will usually be a slightly diffusing duct which will not penalize either take-off or high-speed performance. The ducts should have very smooth inner surfaces, gradual diffusion, and no sharp changes in section. The inlet should be designed to prevent flow separation at applicable angles of attack and flight speeds. A flexible connection is required at the engine inlet to provide for engine expansion. On multiengine aircraft where it is desirable to shut down an engine in flight, the inlet duct should include an air shut-off valve to prevent the engine from windmilling, since a windmilling engine appreciably increases the total drag. The ultimate location and design of the inlet duct should be checked to be sure that this section does not cause a serious lowering of the critical Mach number. Gas turbine powerplants consume large volumes of air and appreciable quantities of fuel but they cannot digest solids without ill effects. Precautions *must* be taken to keep foreign matter, including personnel, away from the duct inlet.

■ Engine Suspension

Engine suspension is the next logical installation question to be raised. Fig. 4 shows the three-point mounting system of turbo-jet engines which lends itself beautifully to simple, rugged supports. The front support is a universal which takes vertical loads only. The two rear trunnions located near the center of gravity of the engine take all fore-and-aft, side, and torque loads. One of the mounting brackets must be a slip fit on the trunnion to allow the engine to expand and to prevent structural loads from



■ Fig. 2 – Engine location



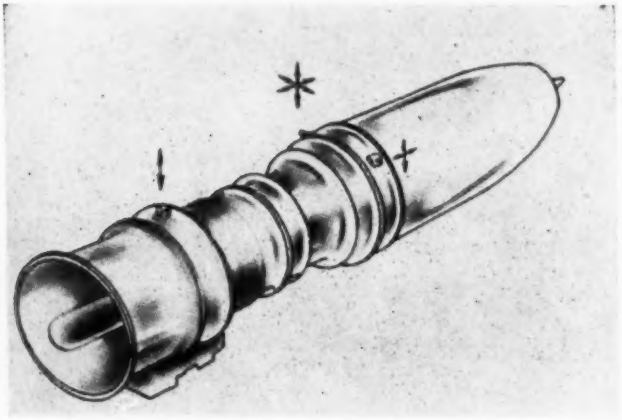
■ Fig. 3 – Air inlet duct

being transmitted through the engine. The front support likewise allows the engine to expand lengthwise and move laterally, thus preventing transmission of airframe loads through the engine.

The vibratory frequency imposed on the engine mounts by turbo-jet engines are usually well above the range of the natural modes of vibration of the aircraft mounting system and structure. In addition, the vibratory loads in all directions are so small that vibration isolators or absorbers are not required. This eliminates the very troublesome problems of providing for engine motion in the design of cowling, inlet ducting, exhaust extensions, ejectors, and so forth. The problem of mounting the unit then consists of a straightforward design to take thrust, torque, gyroscopic, flight, landing, and any other loads peculiar to the intended operation of the aircraft. The simplicity of the mounting trunnions is well adapted to design of quick-disconnecting mounting provisions to facilitate rapid engine changes.

■ Fuel System

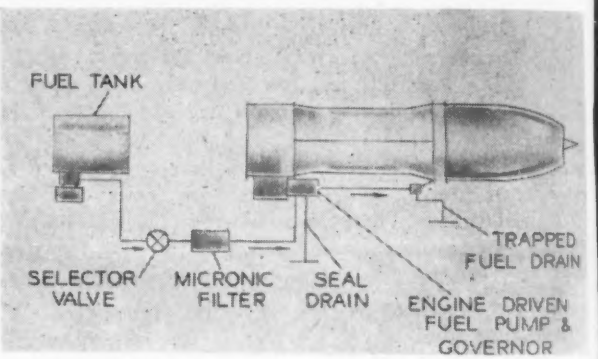
The engine fuel system shown schematically in Fig. 5 includes the high-pressure fuel pump and governor necessary to spray fuel into the combustion chamber at the proper rates. There are two engine connections which must be made in addition to the main fuel line to the engine pump. These are the pump seal drain and the line from the automatic valve which prevents fuel from being trapped in the engine at starting and at shutdown. The aircraft fuel system requires a booster pump capable of high capacity but with the usual booster pump pressure. In addition to these customary requirements the booster pump should have a low pressure drop to permit high-capacity flow if the electric motor is not functioning. High-pressure pumps and governors are sensitive to dirt. It is necessary, therefore, to provide a low pressure drop micronic filter to ensure long, trouble-free life from these accessories. From this it can be seen that the turbo-jet engine fuel system installation does not present any unique problems.



■ Fig. 4 – Mounting system

■ Lubrication System

The requirements of the engine lubrication system are easily met. As pictured in Fig. 6, oil need only be piped to and from the oil pump furnished on the engine. Oil cooling is provided in the engine. Therefore, the piping and a tank which meets the requirements for filling, drainage, expansion space, warming, outlet, return, and vent are the components necessary to complete the oil system.



■ Fig. 5 – Fuel system

Static leakage of oil into the engine airflow path, where it might cause hot starts, is prevented in the engine itself. The entire oil system installation is further simplified since the oil consumption rate is very low. Thus, the small tank can be conveniently located to give the best airplane arrangement.

■ Instrumentation

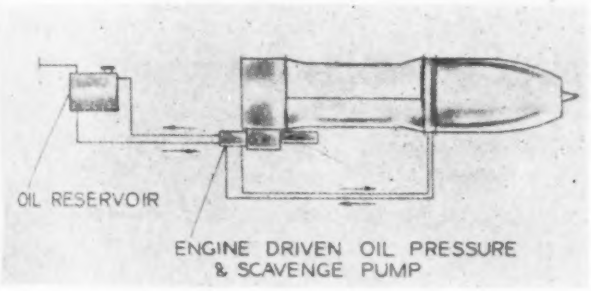
The instrumentation necessary to check proper engine operation is indicated in Fig. 7. The scales on the instruments must, of course, be of the proper range for each temperature, pressure, or rpm of the particular model being installed. This instrumentation is typical of a production airplane and obviously does not include many temperatures and pressures which it might be necessary to obtain during the flight test of an experimental airplane or engine. These instruments are certainly no hindrance in the progress of developing a simple, well-coordinated engine installation.

To ensure proper engine operation, the ambient temperature around the engine accessories should not exceed the allowable limit of ambient temperature for the electrical

components. One satisfactory arrangement for providing cooling airflow is shown in Fig. 8. In this arrangement a pressure drop is maintained across the engine compartment by means of ram in flight and ejector action on the ground. Other schemes accomplishing the desired circulation would be equally acceptable. In addition to cooling the engine, the airframe structure must be protected from engine heat. The curve above the engine is a temperature gradient along the engine surface. From this it can be seen that temperatures high enough to cause material damage to all-metal structure are not encountered until the region of the combustion chamber is reached. From this point aft, it is necessary to protect adjacent structure if the combination of distance from the engine and cooling airflow does not keep the structure below the critical temperature for the material used. The problems involved in making provisions for engine compartment cooling and structural cooling can be solved in several different ways, none of which are excessive in weight or unduly complicated.

■ Engine Control System

The turbo-jet engine control system shown in Fig. 9 consists of two or three controls, depending upon whether



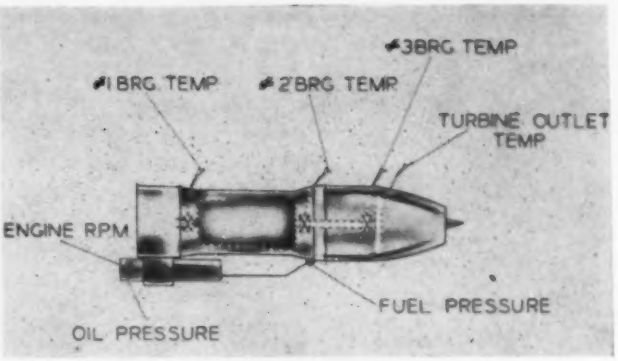
■ Fig. 6 – Oil system

the airplane is single or multiple engine. A single-engine airplane's control system needs only a starter button and a throttle control. Multiengine airplanes require, in addition to the starter and throttle, an air duct butterfly valve, as previously described. This equipment may be located to give the best airplane arrangement. The heavy wires to carry high starting currents naturally should be kept short to save weight. A standard external power receptacle is required for repeated ground starting. Windmill starts are very easily accomplished in flight with normal sized batteries. The engine's performance is controlled by the governor. Each lever setting on the governor is an engine rpm point. Therefore the pilot sets engine rpm when he sets his cockpit control lever to any given position. It can be seen that the turbo-jet engine control system has been designed to meet the requirements of the pilot and to facilitate installation.

This covers the most important of the problems likely to be encountered in turbo-jet installations.

■ Gas Turbine Propeller Drive Engine

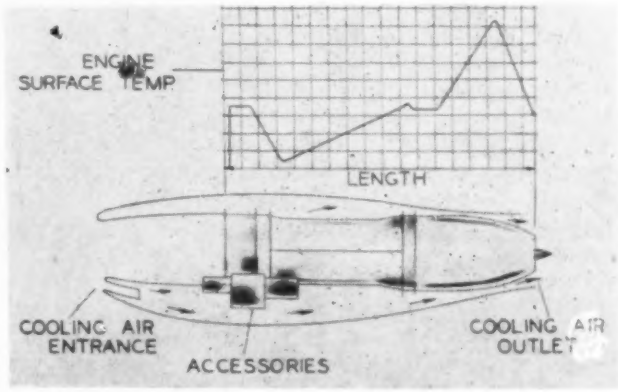
Let us look at the gas turbine propeller drive engine briefly to see what new problems we may expect to encounter.



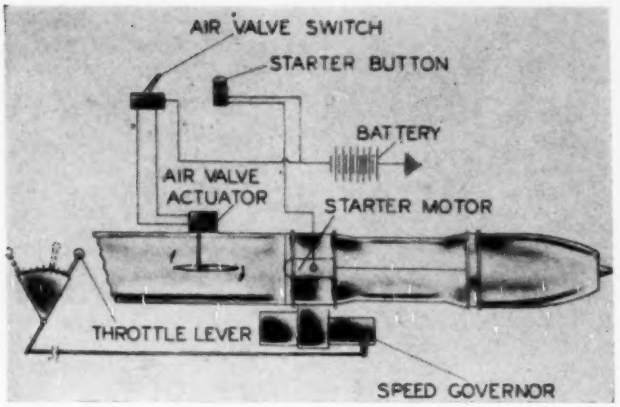
■ Fig. 7 – Instrumentation

The general proportions of the propeller drive, as shown in Fig. 10, are about the same except that the length is increased due to additional compressor and turbine stages and reduction gearing. The flexibility of engine location is reduced, due to the usual propeller clearances required. The induction system problems are similar except that the air is brought in around the outside of the reduction gears. The mounting system must be designed to take the propeller loads in addition to the engine loads, as previously explained for turbo-jet engines. The oil system for the engine is similar to that for a turbo-jet engine but the additional heat generated by the reduction gearing must be dissipated by the oil system. Since this may be several per cent of the propeller shaft power, considerable oil cooling must be provided. This is especially true on the ground with the propeller in low pitch and at high rpm, with resulting low slipstream velocity. Engine cooling requirements are similar to those for a turbo-jet engine, with special care being given to ground cooling. The engine control system, of course, requires a propeller control in addition to the engine speed control. From this brief outline of the turbo-propeller engine installation requirements, it is obvious that additional problems are present. However, forethought is being given to them; and the advantages of small size, high power, and low weight gained with this type of engine will not have to be compromised to make very successful installations.

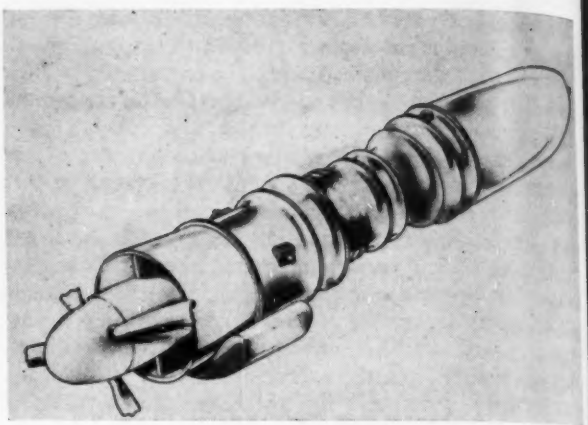
The engine manufacturer is willing to take the responsibility for the design of various parts of the powerplant systems when, by so doing, the resulting engine will con-



■ Fig. 8 – Engine cooling requirements



■ Fig. 9 – Engine controls



■ Fig. 10 – Propeller drive

tribute to the development of better airplanes. From the above description of aviation gas turbine installation problems, it is obvious that this has already been done to a large extent. The small size, light installed weight, and simplicity have been the result of coordination of design to solve the problems of the customer, the airplane manufacturer, and the engine manufacturer best. The future axial-flow gas turbine engines will have even more design features which have been coordinated to solve the engine installation problems of the aviation industry.

Cruising Economy by Use of Water Injection

continued from page 87

25,000 ft, respectively. As the altitude increases, the advantage of using water to implement cooling becomes greater, until at 25,000 ft it is possible to fly 35 mph faster for the same cost by adopting the constant 0.070 F/A ratio setting and adding water.

This is an optimistic picture. However, it unfortunately does not include one factor which, if applied at its full penalty, may make the adoption of water cooling a very unprofitable venture. This is the joker which appears in every new deck, and in this case takes the form of a question. Who pays for the numerous pounds of water which are put aboard to do this engine cooling? Water in itself may not cost a penny but, if the now famous handful of hardware saves a hundred dollars a year per pound to some operators, the water we are forced to carry cannot ride free.

Therefore, we come to our last curve. Fig. 12 illustrates the operating cost if the water we carry displaces an equivalent weight of cargo normally bearing income at the rate of $0.45 a ton-mile. If the operation to be undertaken suffers the full penalty of this payload loss, it can be seen at once that the experiment would not be a success, for at 70% power it will cost $0.80 per mile to use water and only $0.25 per mile with conventional auto lean mixtures. However, it is by no means true that an airplane is operated at all times at 100% load factor, or that the operating range is such that the airplane is limited by weight rather than by volume. Furthermore, some airplanes are limited by landing weight rather than take-off weight, in which case, the water being expendable, no penalty would be imposed.

In addition there has been considerable investigation into the design problems of water recovery equipment. If the drag penalty – in weight and in air requirement – can be reduced sufficiently, water recovery from the engine exhaust may be the answer to the severe cost penalty of carrying water.

In conclusion, therefore, it cannot be stated that water injection during cruising operation offers an economic advantage of a definite number of cents per mile. It does offer, however, a substantial increase in airplane speed under certain conditions without any accompanying increase in fuel cost. It remains for later studies to evaluate, first, the exact penalty of the weight of water carried and, second, the economic value of the increase in air speed. Even operating with normal mixture settings there has been a question in the mind of some operators as to the possible value of cruising at powers in excess of 70% rated. If this procedure is not considered out of the question now, it should be considerably more attractive with the use of water.

Lastly, it will be necessary for some group – after the slide rule has exhausted its potentialities – to fill an extra tank with water and prove the truth or error of these calculations in the only way a controversial question can be answered in aviation: *in flight.*

ADDITIONAL REFERENCE

NACA ARR No. 3K09 (1943), "Effect of Water Injection on Cooling Characteristics of Pratt & Whitney R2800 Engine," by R. J. Koenig and G. Heiser.

Refrigeration for Air Conditioning Pressurized Transport Aircraft

by BERNARD L. MESSINGER
Group Engineer, Thermodynamics
Lockheed Aircraft Corp.

THE need for refrigeration in aircraft air conditioning has not been generally recognized in the past.

Shown here are the reasons why refrigeration is a necessity. Three different methods of accomplishing it are also presented.

The cooling requirements of the Lockheed Constellation are used as a basis for the study.

THE AUTHOR: BERNARD L. MESSINGER, head of the thermodynamics group of Lockheed Aircraft Corp., has been with the organization since 1940. Starting in the research laboratory as mechanical research engineer, he later moved to the design section of the engineering department, acting in a staff capacity as a mechanical and hydraulics design engineer. Mr. Messinger is a graduate of Illinois Institute of Technology.

THE Lockheed Constellation was the first transport aircraft to include refrigeration equipment as part of its air conditioning system. This system, which was conceived as part of the initial airplane design in 1939, also pioneered the application to aircraft of such features as the recirculation fan and heated cabin wall panels.

The original refrigeration equipment was of the vapor compression type and was designed and developed by Lockheed. This development was interrupted by the United States' entry into the war, at which time the Constellation airplane underwent numerous design changes to adapt it to the requirements of a military transport. In the course of these changes the refrigeration equipment was deleted.

In the early part of 1945, when the airplane design was again being revised, this time to conform to its original purpose as a commercial airline transport, it was decided to attempt to make it the most comfortable airplane that would be available to the airline industry. All of the original air-conditioning features, as well as several new ones, have, therefore, been incorporated into an entirely new system. The scope of the operating conditions has been increased to meet every conceivable requirement that the airline operators have voiced. The result is a very complete and yet flexible system.

[This paper was presented at the National Air Transport Engineering Meeting, Chicago, Dec. 5, 1945.]

A design analysis of the entire air conditioning system is beyond the scope of the present paper. Although the heating and ventilating functions involve several novel and interesting arrangements of equipment and ducting, it was felt that a study of the refrigeration function would be of particular interest, since it is an entirely new problem and involves the use of equipment which has been developed expressly for this application.

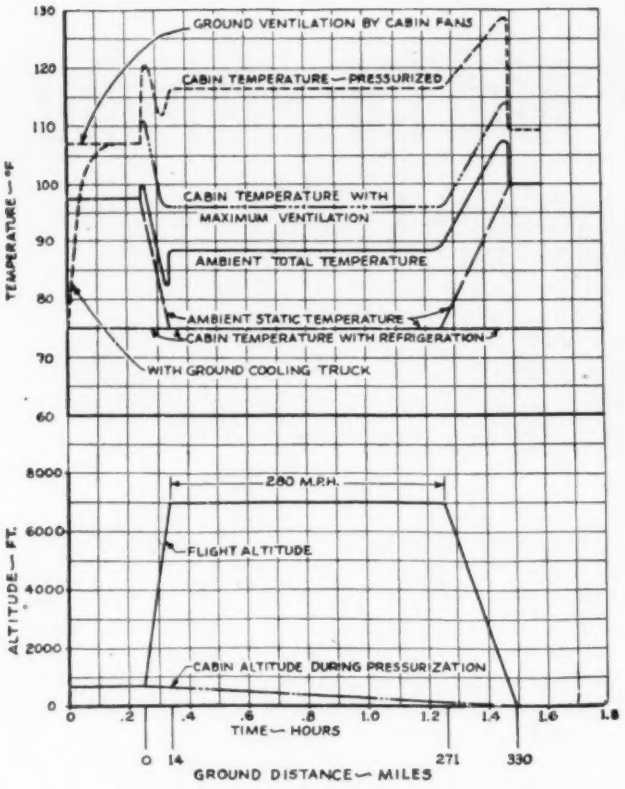
Experience gained during the development of the vapor cycle refrigeration equipment indicated that it would be necessary to devise some new method of refrigeration for the rather severe requirements of an aircraft application. Partly as the result of related wartime development programs, it became apparent that an air-cycle refrigeration system could be designed for a very small expenditure of weight and space. The new air conditioning system, therefore, will employ an air-expansion-turbine unit in place of the vapor-compression unit which was included in the original version of the airplane.

It is the primary purpose of this paper to justify the selection of the air cycle as the most practical type of refrigeration for application to pressurized transport aircraft. The air cycle will be compared with two other possible methods. The use of the Constellation airplane for the purpose of illustrating the application of these methods is incidental. In the opinion of the author the conclusions are applicable to all pressurized transport aircraft of comparable size and service.

■ Significance of Comfort

The aircraft industry has just recently reached that stage in its development where the subject of passenger comfort is receiving a degree of attention that is comparable to that which was formerly devoted entirely to the external performance of the airplane. Prior to this time the aircraft industry had assumed that the airline traveling public would be content to accept a certain degree of discomfort in exchange for faster transportation. The recent change in viewpoint stems mainly from the recognition of the fact that the majority of the potential airline customers will no longer be willing to make this concession.

In other words, the traveling public which has been exposed to comfort features, such as air conditioning in the competitive forms of transportation, is now going to expect the same of airline transportation. This latter fact



■ **Fig. 1 – Temperature and altitude plotted against time, Constellation flight from Los Angeles (706 ft) to San Francisco (6 ft) – 330 miles, Army summer day**

is particularly significant because airline transportation has been represented to the public as a superior means of travel and any discrepancy in this concept due to discomfort factors will certainly have an unfavorable effect on the growth of its acceptance. An excellent analysis of this phase of the problem has been made by McFarland.[1]

Just to review briefly the various factors related to passenger comfort, the following items are listed:

(a) Temperature.
(b) Pressure (or cabin altitude).
(c) Rate of pressure change (or rate of climb and descent).
(d) Noise.
(e) Vibration.
(f) Air movement (or distribution).
(g) Ventilation (or oxygen supply).
(h) Humidity.
(i) Odor, dust, and smoke.
(j) Airplane attitude (angle and bumpiness).
(k) Fixed equipment (seats, berths, lounges, lighting, and the like).

If the cabin pressurization equipment is considered a part of the air conditioning system, it can be seen that the latter exerts control over all of the comfort factors except (d), (e), (j), and (k).

In the case of item (j) the bumpiness can be diminished by making use of the cabin pressurization equipment and flying the airplane in the relatively smooth air that prevails at high altitude. Where short-range, hot-weather operations are involved and the optimum cruising altitude is below 10,000 ft, the air is likely to be very bumpy. Flights under these conditions will benefit greatly from a cabin cooling system which will tend to suppress airsickness by its psychological as well as physiological effect.

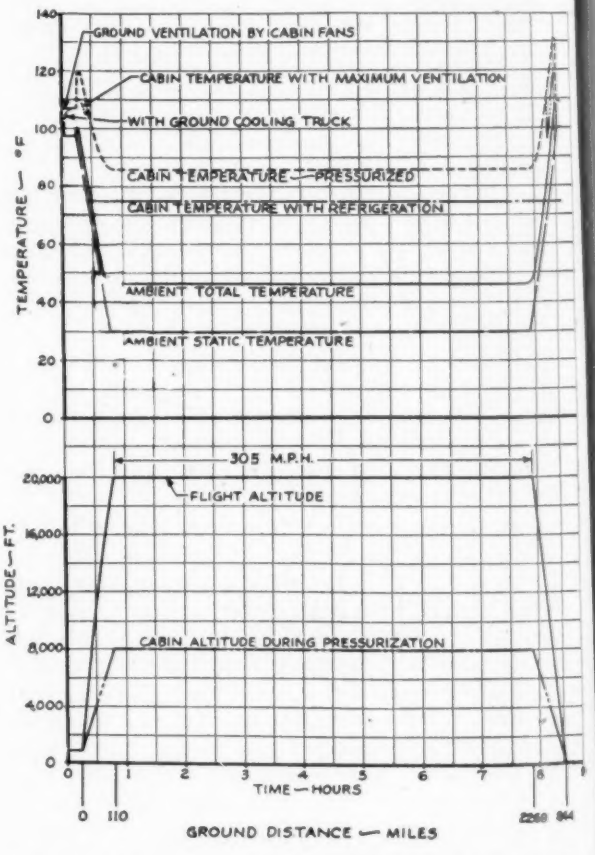
The control of noise and vibration, items (d) and (e), is receiving a great deal of study by all of the large airframe manufacturers.

It is thus apparent that the overall control of passenger comfort depends on a variety of factors. Refrigeration, the subject of this paper, influences only two of these, namely, temperature and humidity, and yet, without adequate control of these two, the significance of the remaining items is greatly diminished.

■ Need for Refrigeration

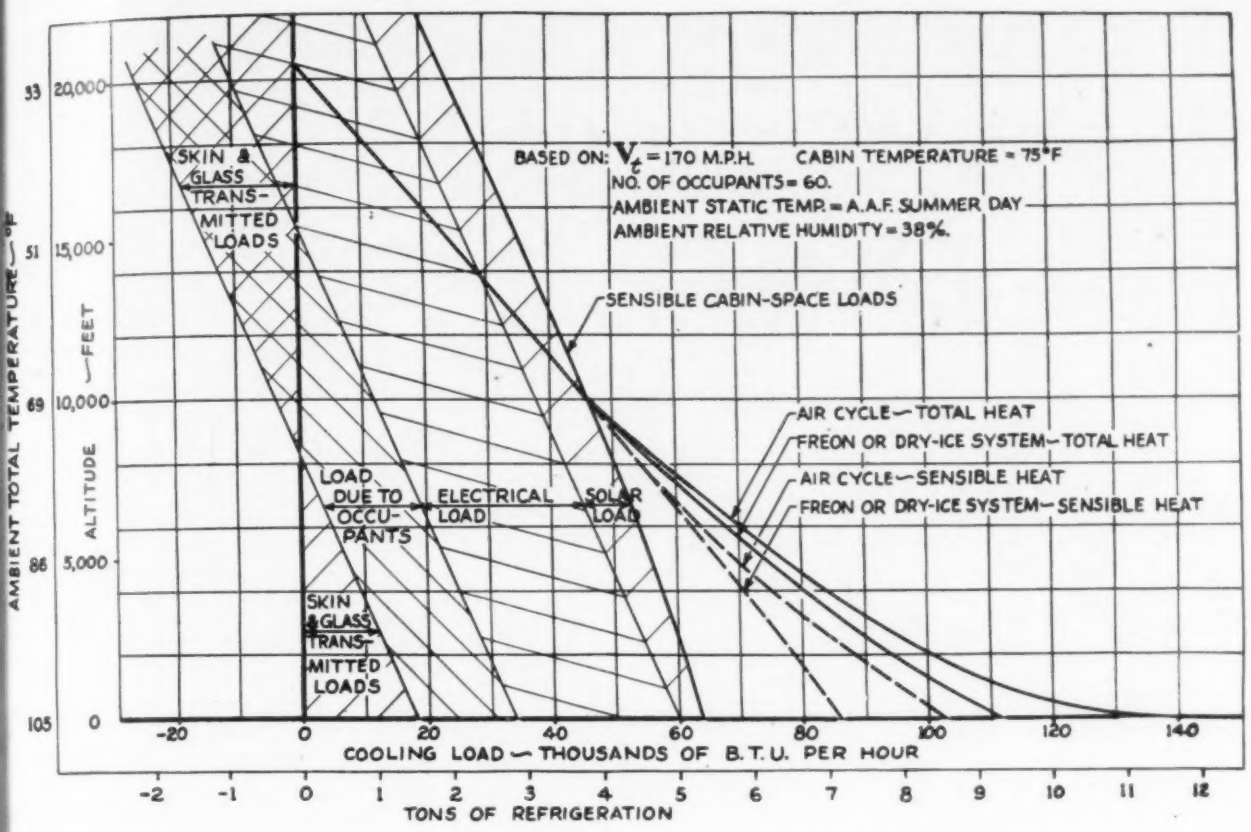
It is a common misconception among many engineers that aircraft air conditioning does not require refrigeration simply because airplanes can be flown at a sufficiently high altitude to avoid excessive ambient temperatures. That this is only approximately true for unpressurized aircraft and not at all the case for pressurized transports can be rather easily demonstrated.

The principal fault in the above concept is that on a design summer day, based for example on the Army maximum temperature-altitude relationship, a 75 F ambient temperature is not attained until the airplane has reached an altitude of 7000 ft. The existence of a 75 F ambient *static* temperature does not mean that the cabin can be



■ **Fig. 2 – Temperature and altitude plotted against time, Constellation flight from Los Angeles (706 ft) to New York (12 ft) – 2440 miles, Army summer day**

[1] See "Factors in Design of Air Transport Planes Influencing Comfort and Safety in Flight," by R. A. McFarland. Division of Research, Graduate School of Business Administration, Harvard University, 1945.


BASED ON: V_t = 170 M.P.H. CABIN TEMPERATURE = 75°F
NO. OF OCCUPANTS = 60.
AMBIENT STATIC TEMP. = A.A.F. SUMMER DAY
AMBIENT RELATIVE HUMIDITY = 38%.
SKIN & GLASS TRANS-MITTED LOADS
SENSIBLE CABIN-SPACE LOADS
AIR CYCLE — TOTAL HEAT
FREON OR DRY-ICE SYSTEM — TOTAL HEAT
AIR CYCLE — SENSIBLE HEAT
FREON OR DRY-ICE SYSTEM — SENSIBLE HEAT
LOAD DUE TO OCCU-PANTS
ELECTRICAL LOAD
SOLAR LOAD
SKIN & GLASS TRANS-MITTED LOADS
AMBIENT TOTAL TEMPERATURE — °F
ALTITUDE — FEET
33 20,000
51 15,000
69 10,000
86 5,000
105 0
-20 0 20 40 60 80 100 120 140
COOLING LOAD — THOUSANDS OF B.T.U. PER HOUR
-2 -1 0 1 2 3 4 5 6 7 8 9 10 11 12
TONS OF REFRIGERATION

■ Fig. 3 – Cooling loads for Constellation cabin during climb

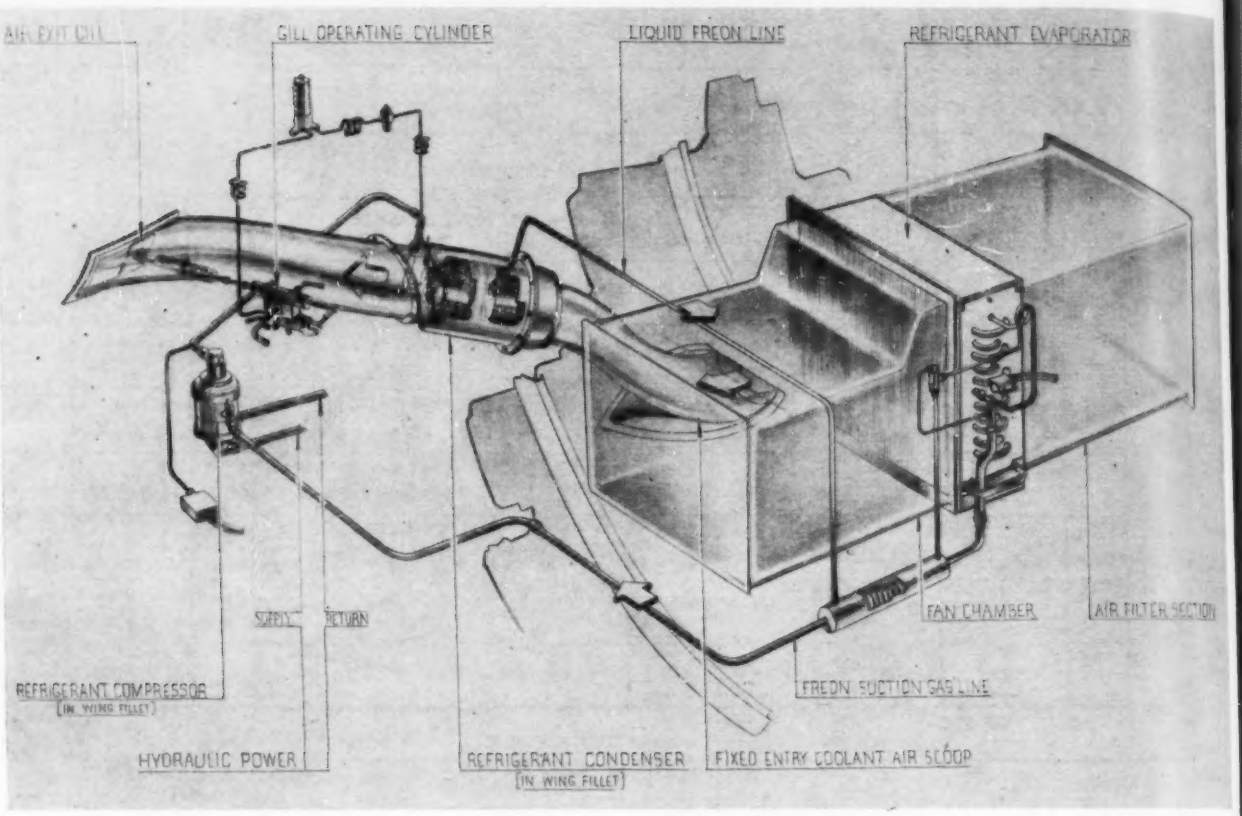
maintained at 75 F without the removal of heat. There are two principal reasons for this. First, there are numerous internal sources of heat in the cabin; among these are the occupants, the solar radiation through transparent surfaces, the various electrical devices, such as radio equipment, inverters, motors, lights, and a slight effect due to thermal lag. The second reason has to do with the difference between static and total temperature or, stated another way, the temperature rise due to stopping (relative to the airplane) air having an initial velocity equal to that of the free stream. For the older type of transport which has a climb speed of 110 to 120 mph, this latter factor amounts to only 2 F, but for present-day aircraft it may amount to 5 or 6 F during climb. In level flight it can become 15 or 20 F. As an interesting sidelight it might be well to point out that in recent high-speed military fighters this temperature rise reaches 50 F or 60 F in level flight. It should be noted that this effect is present not only in the air that is stopped while being brought into the airplane but also in the boundary layer which envelops the entire external surface. The net practical effect is equivalent to placing the airplane in an atmosphere at a temperature very nearly equal to the total temperature.

Due to the above factors it is generally not possible to maintain the cabin temperature within less than 10 F to 20 F of ambient static temperature, even though adequate (one complete air change per minute) ventilation flow rates are provided. Thus at 7000 ft altitude on a summer day the cabin temperature would range between 85 F and 95 F. Obviously, at altitudes below 7000 ft even higher cabin temperatures result. The relationship is not linear, however, because the sensible heat liberated by the occupants diminishes as the cabin temperature approaches body temperature, 98.6 F.

Although of minor significance, it should be noted that thermal insulation in the fuselage wall is a disadvantage in any such condition in which the cabin temperature is above ambient temperature since it limits the amount of heat which can flow outward.

One very significant aspect of the airline operator's problem involves the elapsed time between the terminal and the point of take-off, as well as the corresponding ground maneuvering time after landing. This time element is quite variable. For a 2-engined transport the minimum time for taxiing and engine run-up has been estimated at 5 min and the average at about 8 to 10 min. For a 4-engined airplane the average time would probably range between 10 and 15 min. For large airports operating with a heavy flow of traffic the time required to obtain clearance from the control tower has on some occasions resulted in causing airplanes to stay on the ground for as much as 30 min after leaving the terminal. From the standpoint of the psychological effect on the passenger, particularly a new airline customer, the conditions which prevail in the cabin at the beginning and end of a flight are of primary importance. If a passenger is subjected to discomfort factors such as high temperature, high humidity, poor ventilation, excessive noise and vibration, it is quite likely that the resulting unfavorable impression he receives during the first hour that he is in the airplane will color his overall reaction to the whole flight.

The foregoing discussion, which applies only to unpressurized aircraft, indicates the fallacy in the contention that refrigeration is unnecessary. In the case of airplanes



■ Fig. 4 – Original Constellation refrigeration installation

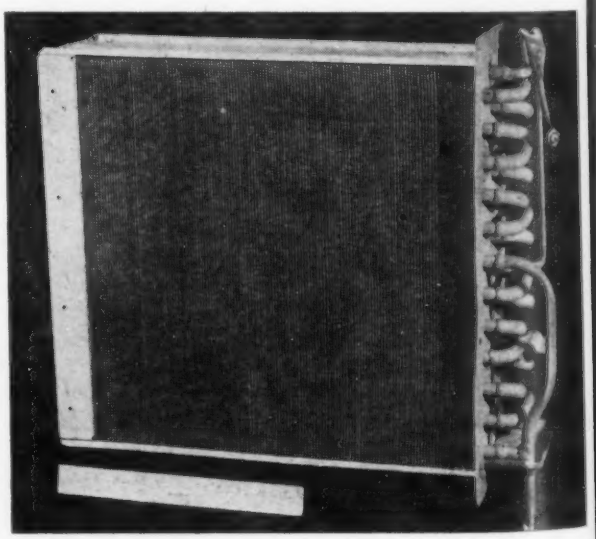
in which pressurization begins at take-off, all of the above considerations are applicable in addition to those contributed by the process of supercharging the cabin. The effect of the latter is due to two factors: (1) the elimination of the large ventilation flow rates as a result of the practical limitations of the superchargers, and (2) the fact that even when aftercoolers of high effectiveness are used, the air enters the cabin at a temperature somewhat higher than the ambient total temperature.

To illustrate the effect of providing an airplane with refrigeration equipment, Figs. 1 and 2 are shown. These two figures are based on two typical flight operations that will be encountered by the Constellation.

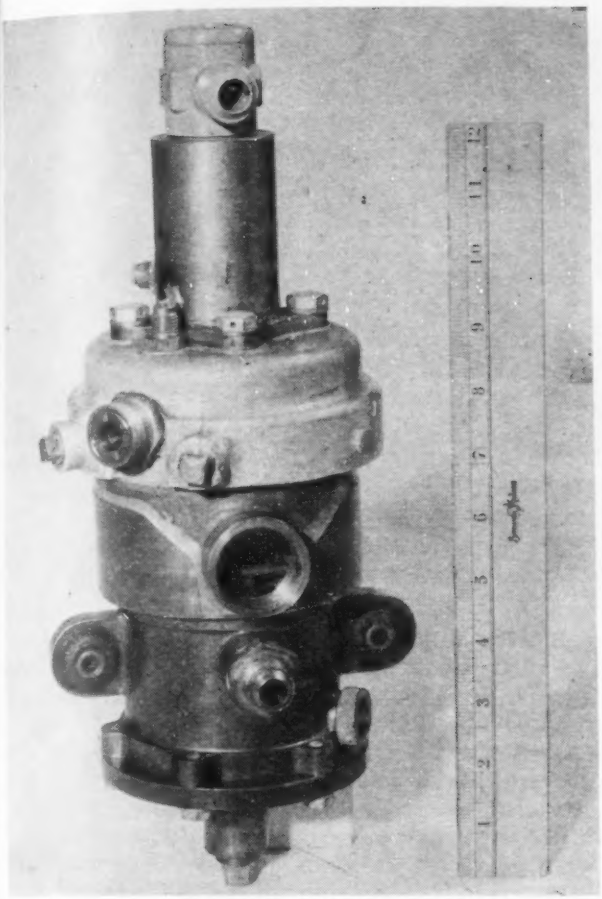
Fig. 1 is plotted for a short-range operation in which the optimum flight altitude is 7000 ft. The altitude schedule is shown as a solid line in the lower portion of the graph. In the upper part all of the significant temperatures are plotted for (1) pressurized flight using intercoolers having 90% effectiveness, (2) unpressurized flight using maximum ventilation rates, and (3) pressurized flight using refrigeration. In a flight at this moderate altitude the temperature conditions in flight are almost as severe as those on the ground. Pressurized flight without refrigeration would obviously be intolerable and flight with maximum ventilation would be little better. For this type of operation the use of refrigeration contributes a very significant degree of comfort.

Fig. 2 is a similar plot for a nonstop transcontinental operation. For this condition a large proportion of the total elapsed time is at an altitude at which cooling by refrigeration is ordinarily not required. On an Army summer day, however, it is apparent that even at 20,000 ft some cooling is required in order to bring the cabin temperature from 86 F down to 75 F. Because unpressurized commercial operation is possible only below 12,000 ft and preferably below 8000 ft, the use of the ventilation system is shown for the ground maneuvering period and the period required to climb to 8000 ft. If the climb is made in the unpressurized condition, the cabin rate of climb is equal to that of the airplane and may cause passenger discomfort unless the airplane flight plan is governed to avoid the latter.

During the ground maneuvering period of both flight



■ Fig. 5 – Refrigerant evaporator



■ Fig. 6 – Refrigerant compressor

plans the only alternative to refrigeration which has been indicated is the use of the cabin fans as a means of ventilation. It would be entirely impractical to introduce supercharged air into the cabin in the absence of refrigeration while on the ground since the low effectiveness of the aftercoolers during this period would result in a cabin temperature of at least 130 F.

■ Fig. 7 – Refrigerant condenser – front view

The foregoing discussion clearly demonstrates why airplanes are as much in need of refrigeration as any of the other forms of transportation.

The design conditions for the Constellation were established after the manufacturer and the airline operators had jointly reviewed the significant operating requirements which would be met in service.

The resulting requirements which have been used are:

1. The airplane refrigeration system is to be capable of delivering the required capacity in flight or on the ground as long as both outboard engines are operating at 1200 rpm or more.

2. A ground truck or its equivalent is considered acceptable for refrigeration of the airplane while it is parked without engines running. The truck is to be as simple and free of service problems as possible.

3. The refrigeration equipment is to be quickly removable as a unit so that during winter operation a negligible weight penalty will be charged to this system. This provision also permits easy access to the equipment for maintenance purposes.

4. The design ambient temperature is to correspond to the Army summer day plus the adiabatic rise.

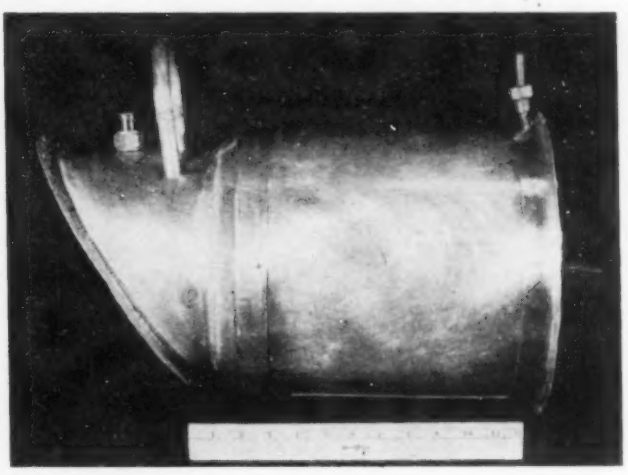
5. An absolute humidity of 109 grains per lb of dry air is the design value at sea level. This corresponds to a relative humidity of 38%, which was considered constant for all altitudes up to 20,000 ft.

6. The minimum design cabin dry bulb temperature is to be 75 F and the cabin absolute humidity 78 grains per lb of dry air or 60% relative humidity.

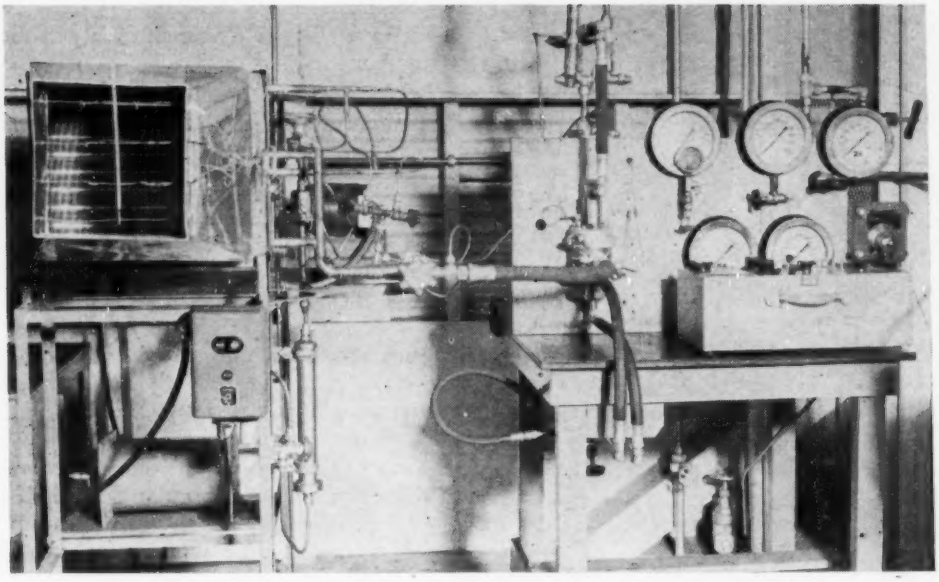
7. The design calculations are based on carrying 60 occupants, although the normal number for the commercial version of the airplane is 54 including a crew of 6. One alternate arrangement provides for 64 passengers.

8. The minimum supply of outside air is to be 50 lb of air per min, which is based on a consideration of the odor and smoke control problems.

Fig. 3 shows the variation in cooling load with altitude during climb. The components of the cabin space load are indicated by the various shaded areas. The convergence of the overall load curves at 10,000 ft is the accidental result of the fact that at this altitude the required airflow rate through the superchargers is the same for all three refrigeration methods. Furthermore, the latent load be-



■ Fig. 8 – Refrigerant condenser – side view



■ Fig. 9 – Experimental laboratory test setup

comes zero at this point and the temperature of the air leaving the primary intercooler is equal to the cabin temperature so that the total cooling load is equal to the space load.

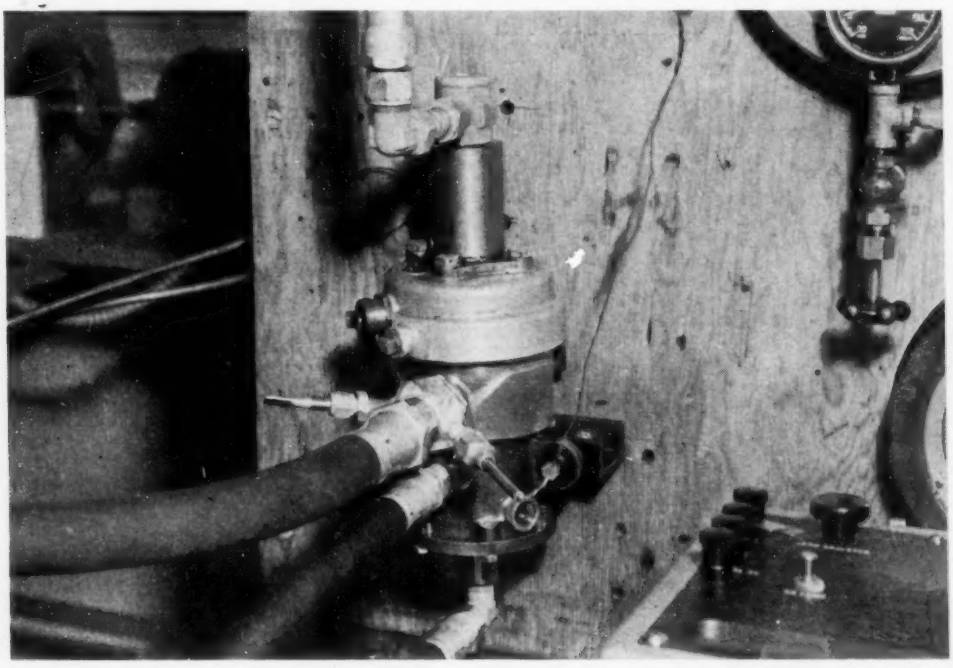
■ Comparison of Methods

Vapor Cycle – The vapor-compression cycle is used in practically all commercial refrigeration applications. The only significant exceptions are the two vapor-absorption cycles, namely the large ammonia-water systems which are generally used where adequate sources of waste heat and cooling water are available and second, the Electrolux cycle which is limited almost entirely to the small loads encountered in domestic refrigerators. These latter applications, even when considered together, represent only a very small proportion of the total refrigeration capacity in use at the present time.

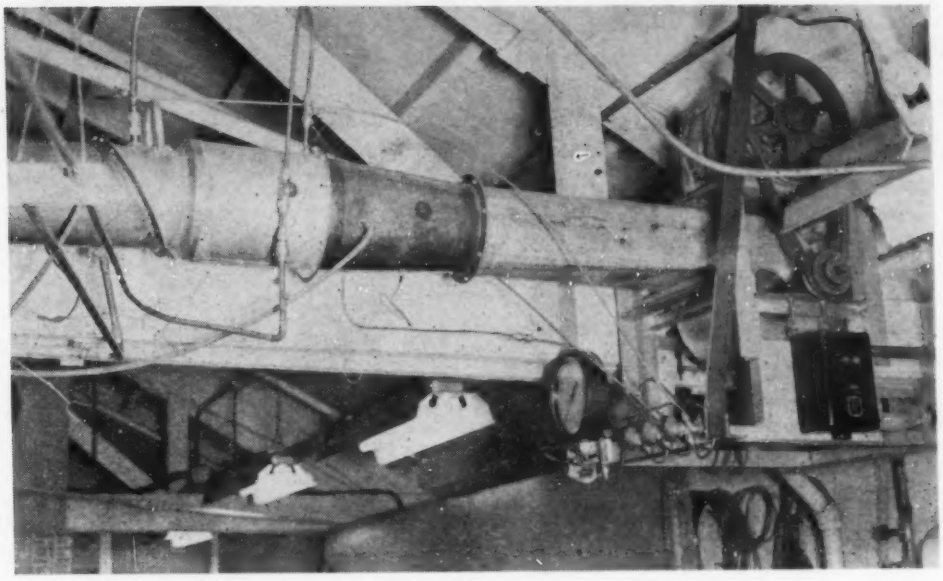
In terms of the energy consumed per unit of refrigerating effect, the vapor-compression cycle is by far the most efficient. It was because of this well known fact that Lockheed's initial refrigeration development program was directed toward this type of equipment.

There are three essential parts to a vapor compression system: (1) the evaporator in which refrigerant is evaporated at the expense of heat which is removed from the air being cooled, (2) the compressor which pumps the vaporized refrigerant from the low-pressure evaporator to a higher pressure at which it can be condensed by ambient temperature air in (3) the condenser which rejects the heat of refrigeration and compression to the surrounding atmosphere.

Fig. 4 illustrates the installation of the original experimental refrigeration system. The three basic elements are evident in the sketch. The evaporator or cooling coil



■ Fig. 10 – Compressor mounted in test setup



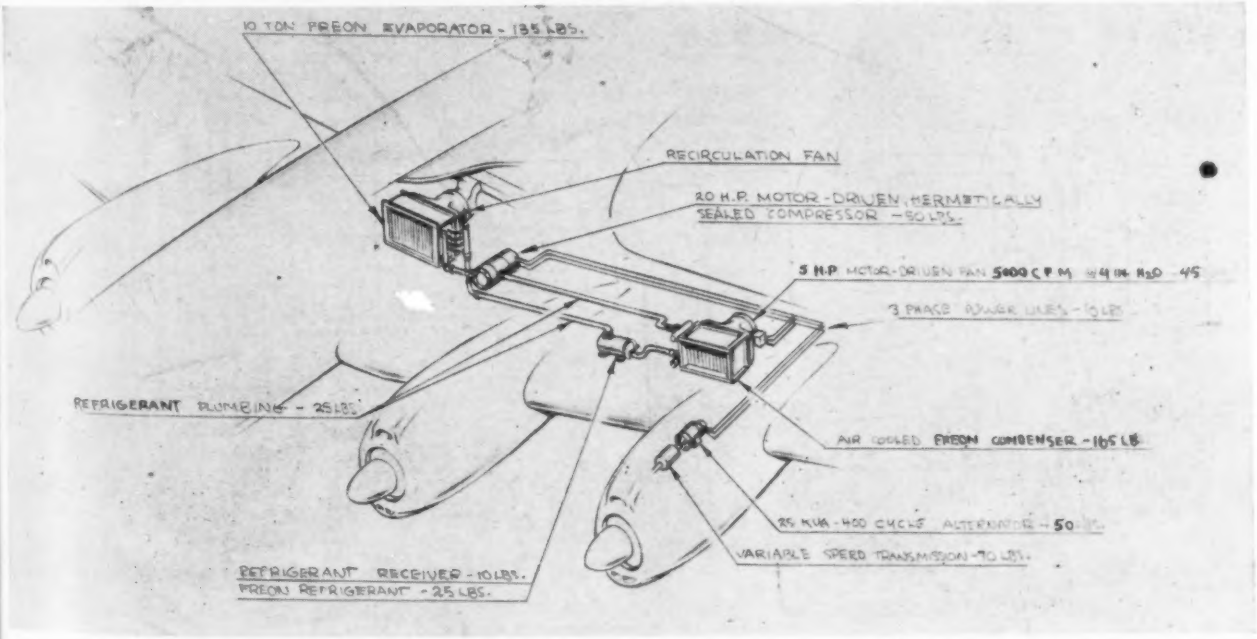
■ Fig. 11 – Condenser mounted in test setup

(Fig. 5) was located in the cargo compartment below the cabin floor. Air was drawn through the coil by a blower located in the box to the left of the coil, after having passed through air filters mounted in the section upstream of the coil. After leaving the blower, the air is mixed with outside air from the superchargers and then distributed throughout the cabin.

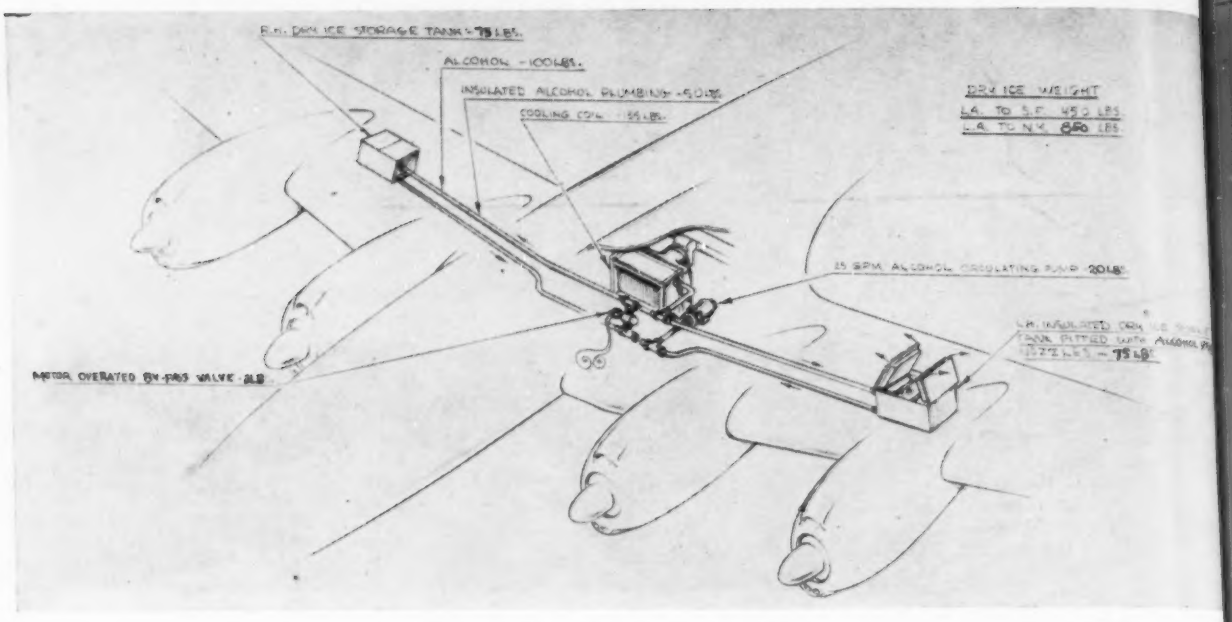
The compressor and condenser were both located in the aft wing fillet just outboard of the fuselage. The compressor, shown in Fig. 6, was a high-speed reciprocating type, powered by a hydraulic motor designed as an integral part of the unit. The refrigerant, Freon-12, was pumped by four double-acting pistons, each of which was directly connected to a corresponding single-acting hydraulic motor piston, the rod connecting the two pistons being fitted with chevron seals for both refrigerant and hydraulic fluid. The only rotating element was a rotary valve in the center of the unit which was driven by a wobble-plate at the lower end of the motor. The valve shaft served for both compressor and motor.

The condenser (Figs. 7 and 8) was built of aluminum tubing having integral spiral fins. The tubing was formed into three helical coils placed concentrically around a cylinder which served as the refrigerant receiver or reservoir. Cooling air was supplied by a duct from a scoop located on the lower surface of the wing root and left the airplane through a thermostatically positioned exit gill in the upper surface of the aft fillet. It should be noted that no condenser cooling air was available except during flight.

Due to the fact that the elements of the system were closely coupled, the Freon plumbing was not very extensive and copper tubing and fittings were used throughout


10 TON FREON EVAPORATOR - 135 LBS.
RECIRCULATION FAN
20 H.P. MOTOR-DRIVEN HERMETICALLY SEALED COMPRESSOR - 90 LBS.
5 H.P. MOTOR-DRIVEN FAN 5000 C.F.M. @ 4 IN. H2O - 45
3 PHASE POWER LINES - 10 LBS.
REFRIGERANT PLUMBING - 25 LBS.
AIR COOLED FREON CONDENSER - 165 LBS.
25 KVA - 400 CYCLE ALTERNATOR - 50
VARIABLE SPEED TRANSMISSION - 70 LBS.
REFRIGERANT RECEIVER - 10 LBS.
FREON REFRIGERANT - 25 LBS.

■ Fig. 12 – Schematic diagram of optimum Freon system


R.H. DRY ICE STORAGE TANK - 75 LBS.
ALCOHOL - 100 LBS.
INSULATED ALCOHOL PLUMBING - 50 LBS.
COOLING COIL - 55 LBS.
DRY ICE WEIGHT
L.A. TO S.T. 450 LBS
L.A. TO N.V. 850 LBS
25 GPM ALCOHOL CIRCULATING PUMP 20 LBS.
MOTOR OVERATED BY-PASS VALVE - 2LB

■ Fig. 13 – Schematic diagram of dry ice system

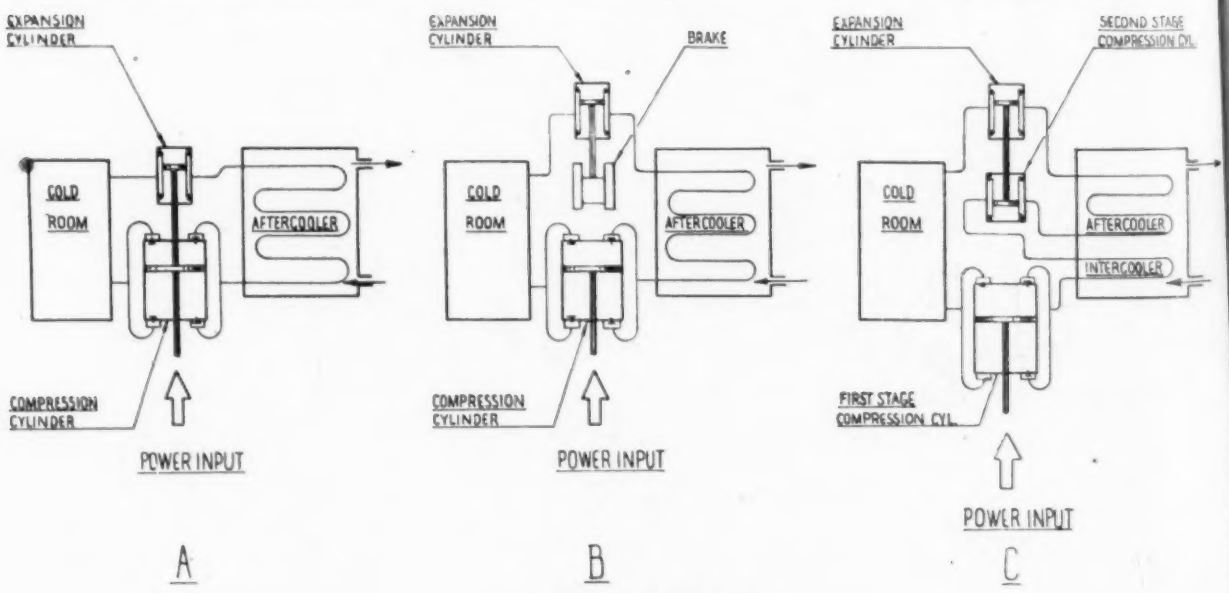
to permit soft solder sweat-joints.

The essential controls are indicated in Fig. 4. The overall output of the refrigeration system was modulated by a throttling valve in the hydraulic supply line to the motor of the compressor. This valve was positioned according to the demand of the main cabin thermostat.

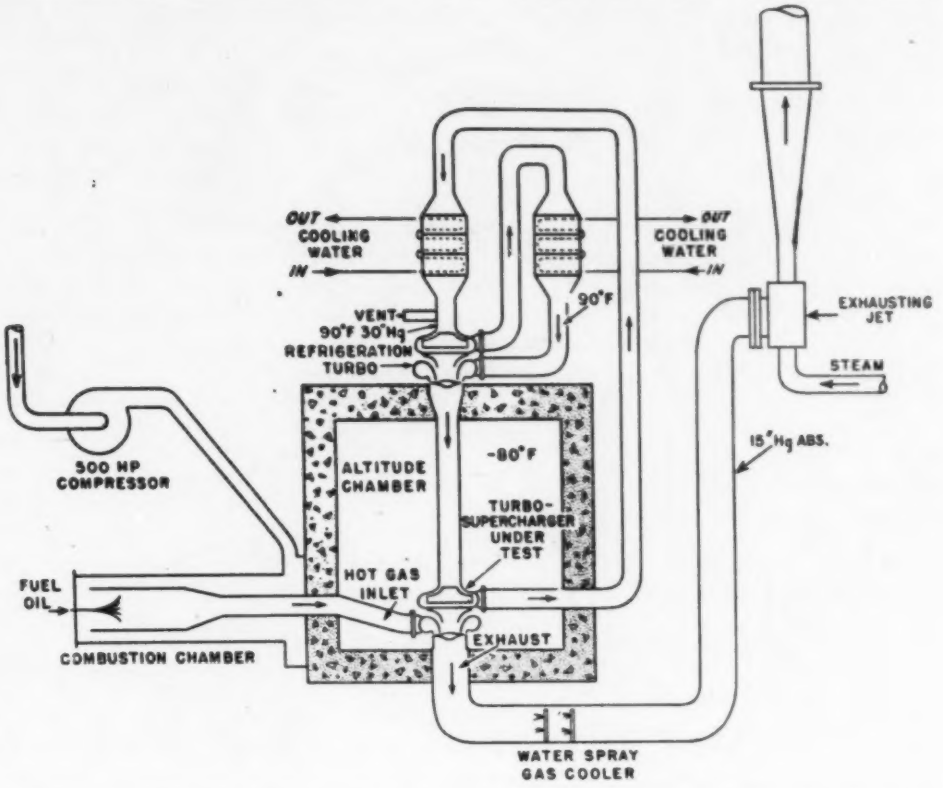
Figs. 9, 10, and 11 are views of the experimental laboratory setup of this complete system. Fig. 9 shows the motor-compressor unit as it was connected to the evaporator at the left. The plumbing shown is not precisely equivalent to that shown in Fig. 4. Fig. 10 is a close-up of the compressor as installed during the test. Fig. 11 illustrates the laboratory installation of the aircooled condenser.

The dimensions of the three basic elements are roughly evident in Figs. 5, 6, 7, and 8 and the corresponding dry weights are 33 lb for the evaporator, 13 lb for the motor-compressor unit, and 16.7 lb for the condenser. The weight of all of the remaining refrigeration components shown in Fig. 4, including the Freon-12 refrigerant, is approximately 35 lb. The total for the system is therefore about 98 lb.

The design operating conditions for the Freon system were 40 F evaporator temperature and 140 F condensing temperature. The capacity of the system by actual test under these rather severe conditions ranged between 1.2 to 1.5 tons, which was admittedly below the moderate requirements to which it had been designed and far below the required capacity indicated in Fig. 3. The evaporator

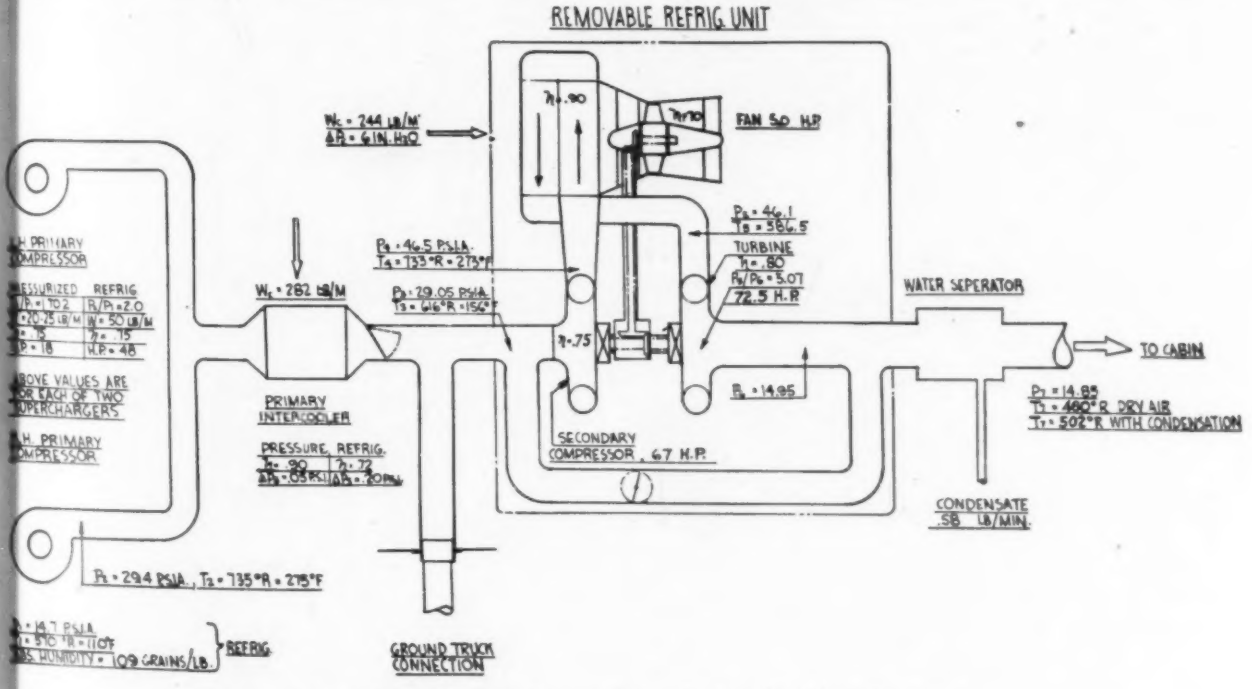


■ Fig. 14 – Air refrigeration cycle

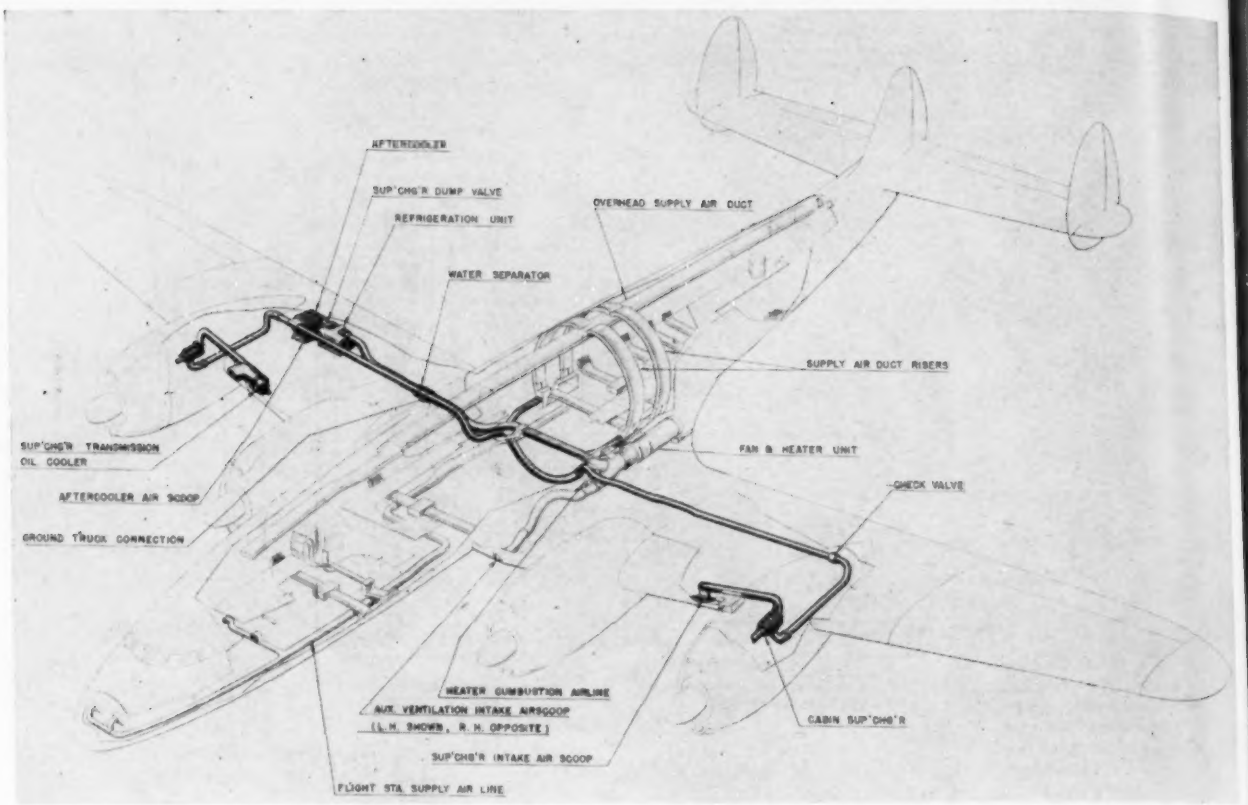


■ Fig. 15 – General Electric air-cycle refrigeration for testing turbosuperchargers – closed refrigeration cycle

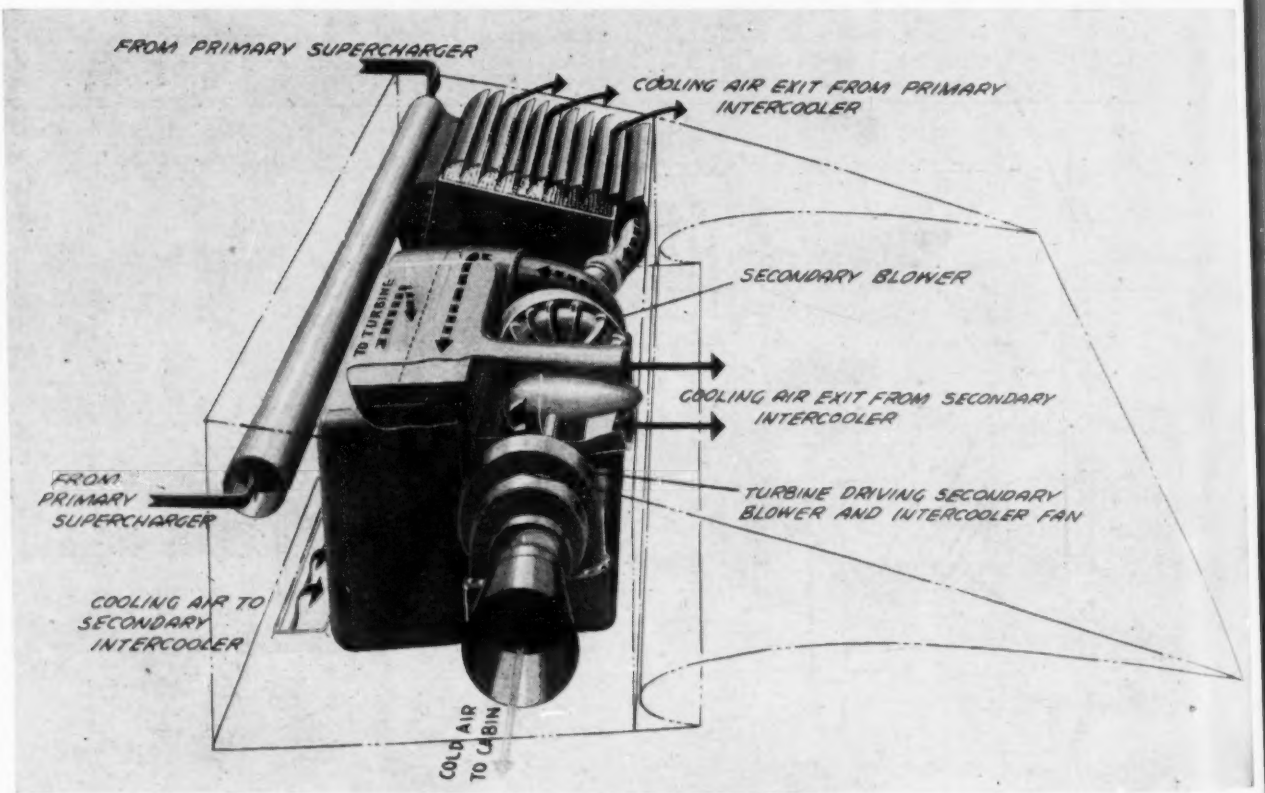
undoubtedly had an ultimate capacity of about 2.5 tons and during the test it was evidently restricted by the limitations of the compressor and condenser elements. An adjusted weight for the system based on an output of 1.25 tons would, therefore, include a deduction of half the evaporator weight or a new total of 82 lb. This corresponds to about 65 lb per ton of capacity, not including any allowance for a portion of the weight of the engine-driven hydraulic pump used to supply power to the compressor. This type of system does not meet the present



■ Fig. 16 – Basic schematic diagram of aircraft air-cycle refrigeration system



■ Fig. 17 – Perspective sketch of Constellation air-cycle installation


FROM PRIMARY SUPERCHARGER
COOLING AIR EXIT FROM PRIMARY INTERCOOLER
SECONDARY BLOWER
TO TURBINE
COOLING AIR EXIT FROM SECONDARY INTERCOOLER
FROM PRIMARY SUPERCHARGER
TURBINE DRIVING SECONDARY BLOWER AND INTERCOOLER FAN
COOLING AIR TO SECONDARY INTERCOOLER
COLD AIR TO CABIN

■ Fig. 18 – Close-up sketch of turbine unit

airline requirement for cooling the airplane on the ground.

In the light of experience gained in the Lockheed development program and also in order to meet present requirements of performance and maintenance, the author is of the opinion that the optimum arrangement of a Freon system would be roughly as illustrated in Fig. 12. To be consistent with the requirements shown in Fig. 3, the nominal rated capacity of this system would be about 10 tons. The three basic components are again evident in Fig. 12 but in this case they are considerably larger than in the early system. Note that the increase in evaporator size consumes valuable additional cargo space. The compressor is of the hermetically sealed type in which the alternating-current driving motor is submerged in the refrigerant atmosphere. Due to the high condensing temperature the power required of the motor is about 17 hp. In order to permit operation while the airplane is not in flight, an electric-motor-driven fan is shown for cooling the condenser. The approximate power required for this fan is 5 hp. Since this particular airplane's electrical system is of the direct-current type, a special alternator of between 20 and 25 kw capacity is shown. It is driven from one of the main engines through a variable-speed transmission. Control of the system's output is accomplished by variation of the speed ratio of the transmission in response to the demand of the cabin thermostat. Allowing for an efficiency of 80% for the alternator and also for the transmission, the power taken from the engine would range between 40 and 50 hp.

Note that this installation does not permit simple access and removability. In order for this requirement to be met, the evaporator, compressor, and condenser would have to be built into a single unit. Although the evaporator and compressor could be combined into a single installation, it is very doubtful that the condenser cooling air supply arrangement would permit the location of the latter inside of the fuselage, to say nothing of the further reduction in cargo space. If a single unit installation were attempted outside the fuselage it would be equally difficult to bring all of the recirculated air from within the fuselage to a single evaporator located outside and then return the air to the inside once more.

The total of the individual estimated weight components shown in Fig. 12 is 585 lb or 58.5 lb per ton of capacity.

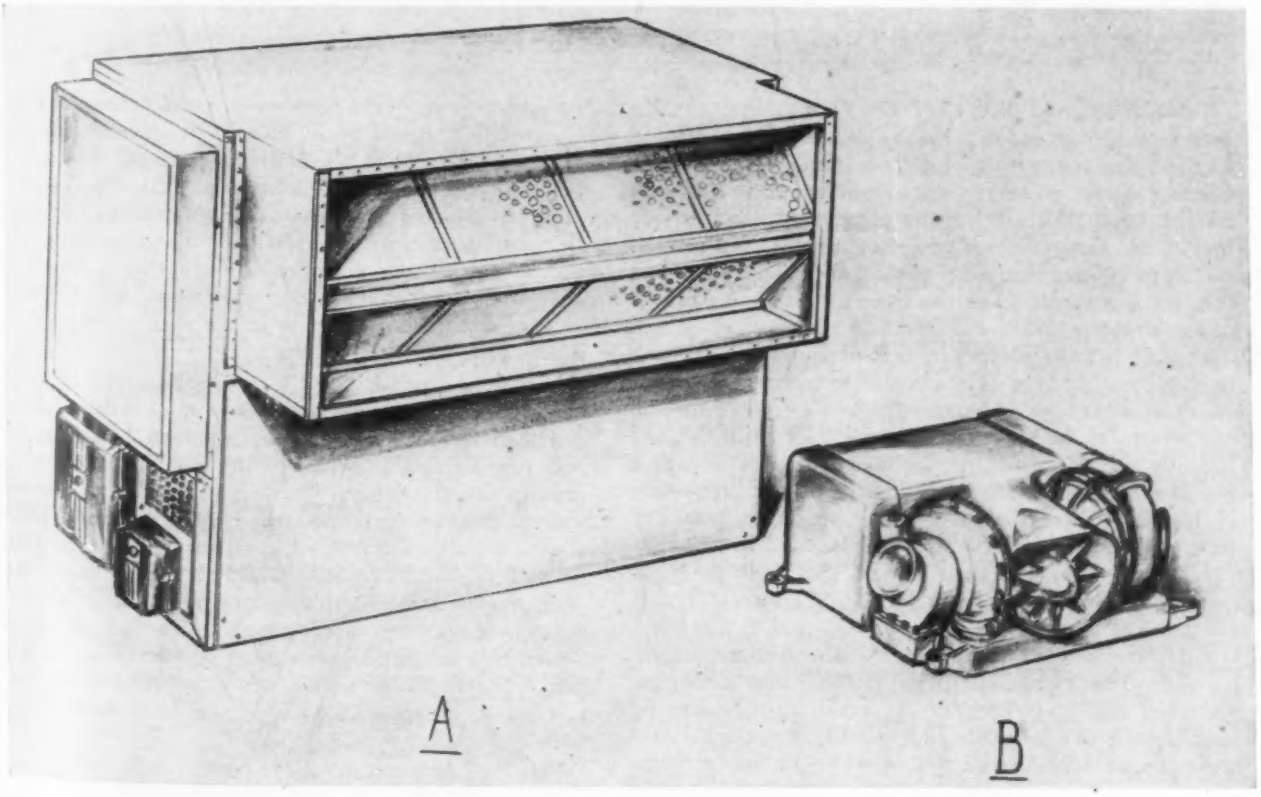
Dry Ice System – The following description of a dry ice system has been included as being representative of the storage method of refrigeration.

The simplest form of this type of system would consist of a means for circulating cabin air over slabs of the frozen material. There are, unfortunately, several reasons why this arrangement would not be acceptable for use in this application. These are:

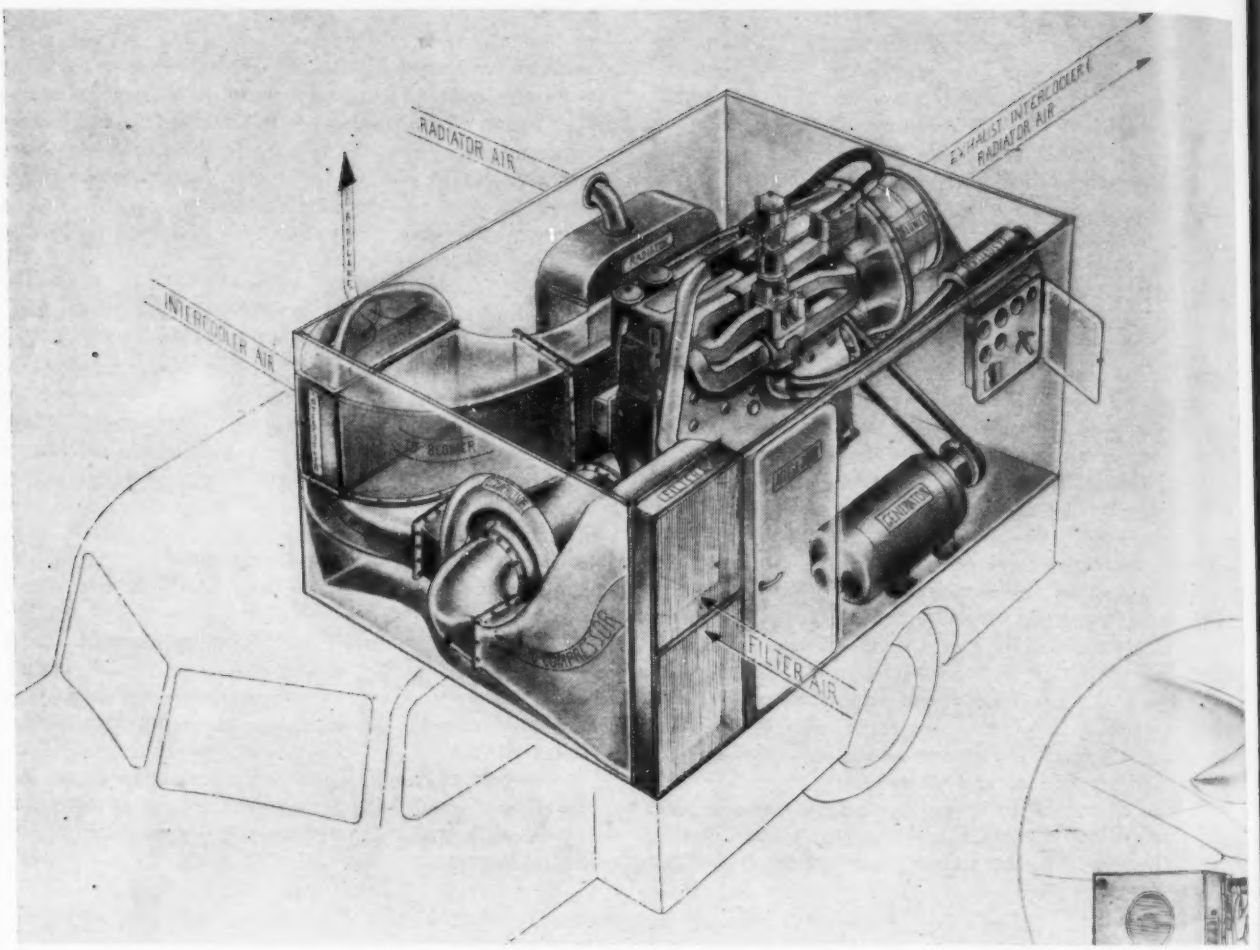
1. The surface area of the ice progressively diminishes during the melting process and thus reduces the heat transfer capacity.

2. Due to the low surface temperature, the condensation of moisture is greater than necessary. In the case of dry ice the very low surface temperature causes the formation of an insulating blanket of frost, thus reducing the rate of heat transfer.

3. The use of dry ice, which has almost twice the refrigerating effect of water ice per unit weight, involves the liberation of a prohibitive volume of CO_2 gas during pressurized operation.



■ Fig. 19 – Comparison of commercial air conditioner *A* and aircraft turbine unit *B*



■ Fig. 20 – Proposed air-cycle ground truck

In order to avoid these objections the simple system has been modified to include a cooling coil practically identical to the Freon evaporator of Fig. 12, and an alcohol circulating system to deliver heat from the coil to a dry ice storage tank. These elements are shown schematically in Fig. 13. The fact that two storage tanks are shown is merely to indicate that the required tank volume including insulation, is greater than the space available on one side of the airplane.

Control is accomplished by a modulating bypass valve in the alcohol system. The circulating pump would operate at constant flow as long as any cooling was required.

The sum of the approximate weights shown in Fig. 13 is 458 lb exclusive of the dry ice itself which would weigh 450 lb for the flight shown in Fig. 1 and 850 lb for that of Fig. 2. Thus, the initial total weight for the Los Angeles to New York flight would be 1308 lb or about 131 lb per ton. The total weight after half the total elapsed time would be about 883 lb or an average unit weight of 88 lb per ton.

From the standpoint of ease of maintenance and removability, this system is at least no better than the Freon system of Fig. 12. There is the additional problem of handling the dry ice while charging the storage tank.

Air Expansion Turbine System – The air refrigeration cycle, which has been selected as most practical for the Constellation, found fairly widespread application during the early days of mechanical refrigeration. Today it is practically extinct in the commercial field. Its disappearance is due to the high degree of development experienced by the more efficient vapor cycle. It is significant that the peculiarities of the airplane are such that a system of refrigeration which has been superseded in every other field of application proves to be the most practical of all in this particular field.

Fig. 14A, which is taken from Goodenough,[2] is typical of similar illustrations to be found in any standard thermodynamics textbook. In this arrangement, known as the open-type air cycle, external power is applied to the compression cylinder which draws air from the cold room at atmospheric pressure and discharges it into an aftercooler which removes the heat of compression. The air then passes through the expansion cylinder which returns the work of expansion to the compression cylinder thus to augment the external power supplied. The heat equivalent of the work done in the expansion cylinder is in this way removed from the air which then enters the cold room at a low enough temperature to produce the required refrigerating effect.

Note that the same amount of cooling could be obtained by dissipating the work of expansion in a friction brake and supplying the work of compression entirely from the

[2] See p. 248 of "Principles of Thermodynamics," by G. A. Goodenough. Holt, New York.

external source of power. This is illustrated in Fig. 14B, and indicates the distinction between a regenerative and a nonregenerative system or, in recent military parlance, between a "bootstrap" and a "nonbootstrap" system.

Fig. 14C shows an arrangement which is functionally equivalent to that of Fig. 14A. In this case, however, the compression process has been divided into two stages, the first of which obtains all of its power from the external source, and the second entirely from the expansion cylinder. Note that intercooling between stages is available in addition to the final aftercooling. Except for the fact that new air, rather than recirculated air, is introduced into the compressor, the Constellation system is basically equivalent to this last arrangement.

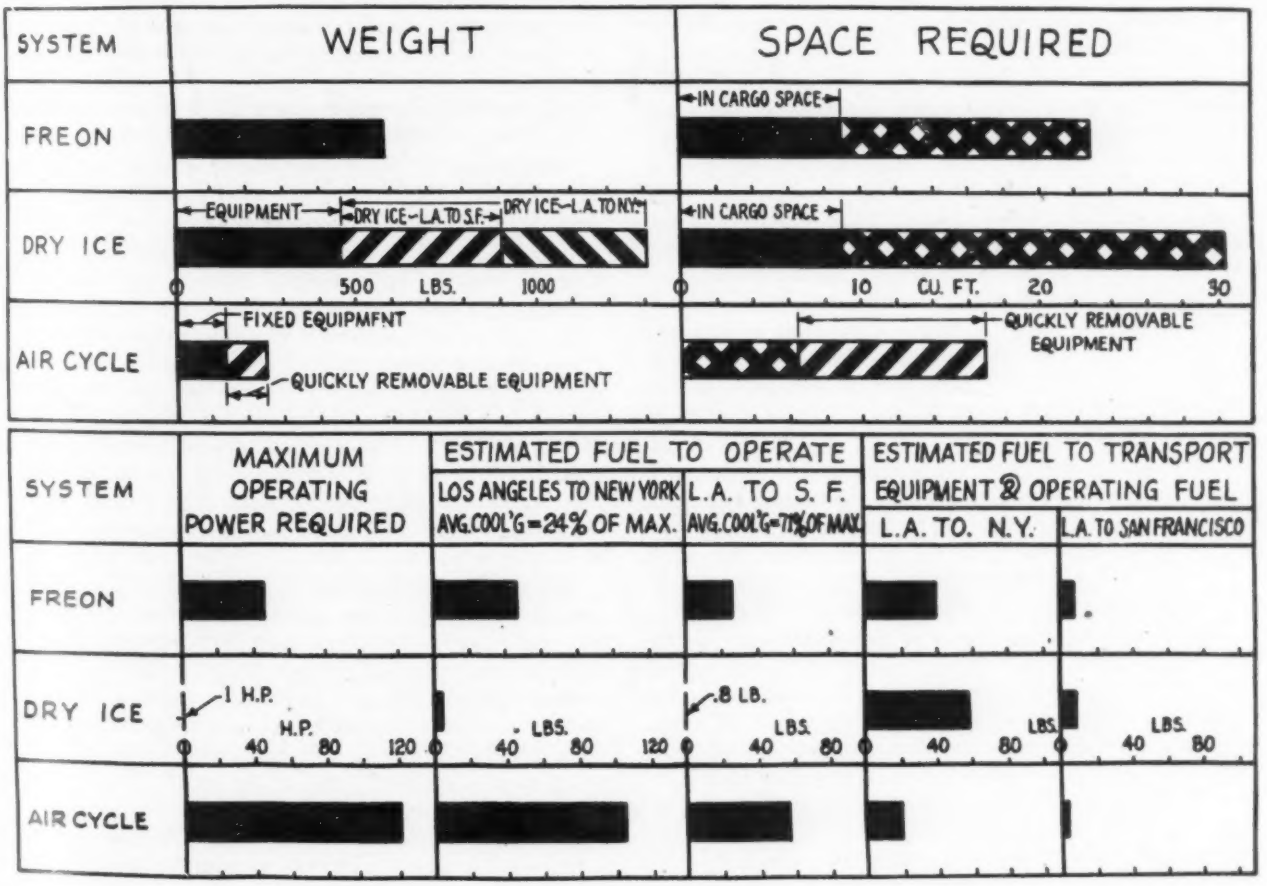
The first application of turbo machinery to the air cycle, that the author is aware of, is described in a book published by GE,[3] from which Fig. 15 has been taken. In this case it is used by the General Electric Co. as a convenient method of obtaining air at a very low temperature for testing turbosuperchargers under high-altitude conditions. When it is realized that a device having the size of a standard aircraft-engine turbosupercharger has a refrigeration capacity of about 100 tons when used as shown in Fig. 15, it is apparent why this method becomes such an attractive solution for the airplane cooling problem.

After Lockheed had reached its decision to install an air turbine system in the Constellation, it was also decided to attempt to procure the necessary equipment from an outside vendor if possible. As the initial step in this program a diagram practically identical to Fig. 16 was submitted to various manufacturers. Of these, the Airesearch Mfg. Co. of Inglewood, Calif., was the only firm which showed significant interest in a program to develop suitable equipment for this application. Considerable credit is due the management and engineering department of this company for their progressive attitude in this cooperative effort.

Fig. 16 illustrates the functional arrangement of the airplane system. The similarity to Fig. 14C is evident. All of the refrigeration equipment is contained in a single removable unit which requires no external connections other than those at the air inlet and outlet. No power is supplied to the unit by mechanical or electrical means. Power enters the system only through the cabin superchargers. These superchargers are no larger in capacity than if designed for pressurization alone. This is due to the fact that they are designed for high-altitude inlet air density and when operated at maximum speed at sea level they have the inherent capacity to supply the increased mass-flow required for maximum refrigeration. Obviously, there is an increase in the power required under this condition at sea level and the transmission design is based on this maximum requirement. All of the significant design quantities are shown in Fig. 16.

In order to meet the airline requirement to be able to cool the airplane on the ground in the absence of ram, the author suggested the solution illustrated in Fig. 16, namely,

[3] See p. 170 of "1943 Supercharger Symposium Lecture Book," General Electric Co., River Works, West Lynn, Mass.



■ Fig. 21 – Comparison of Freon, dry ice, and air-cycle performance and installation factors

the application of the turbine power to both the secondary compressor and the coolant air fan. Prior to this solution it was common practice either to deliver the entire turbine power to the secondary compressor and depend only on ram air for cooling the aftercooler, or to deliver wastefully all the turbine power to an inefficient fan for supplying aftercooler coolant air, the latter fan acting more as an energy absorber than as a fan. That this second method is very inefficient is apparent when the fan power and turbine power shown in Fig. 16 are compared. Because power is not fed "upstream" to a secondary compressor this method has been referred to as the "nonbootstrap" system.

The air-cycle refrigeration machine is very similar to the gas turbine powerplant in one important characteristic. Both machines involve the internal flow of a large amount of power in a "run around" path. Therefore a slight variation in the efficiency of any of these large power handling units has a very critical effect on the overall efficiency of the machine. In the case of Fig. 16 if the efficiency of the secondary compressor and the turbine was in each case 100% instead of 75 and 80% the required primary horsepower would be only 20 hp instead of 120 and the primary pressure ratio 1.2 instead of 2.0. This is the basic reason why the air cycle cannot compete on a power basis with the vapor cycle.

The water separator shown serves to remove about 60% of the entrained free moisture which, under design conditions, is condensed at the turbine. The remaining 40% will be evaporated during the process of mixing turbine air with recirculated cabin air, thereby making this latent cooling capacity available as sensible cooling in the cabin.

Control is accomplished by means of a simple bypass valve as indicated.

The actual airplane installation is more realistically illustrated in Fig. 17. This perspective drawing includes much of the ducting not directly related to the refrigeration system, therefore the latter has been darkened for greater clarity. The air enters the airplane through two flush scoops on the lower surface of each wing leading edge and is delivered to the primary centrifugal compressors which are located in the outboard nacelles. These are driven from the main engines through variable-speed transmissions which permit the full refrigeration capacity to be obtained at engine speeds down to 1200 rpm. Although the transmission is not of the slip-pump type, a small oil cooler is required and is shown in the leading edge just inboard of each outer nacelle.

The discharge air from both compressors is ducted to a single primary intercooler located in the trailing edge section of the right-hand wing. The removable refrigeration unit is located adjacent to the intercooler. From this point an insulated duct takes the cooled air through the water separator and then into the cabin distribution system.

Fig. 18 is a close-up sketch of the installation of the intercooler and the refrigeration unit. It illustrates the very compact arrangement of the various elements. By separating the compressor and the turbine to permit locating the coolant fan between them, Airesearch engineers were able to simplify greatly the problem of driving the fan. A single, right-angle reduction gear transmits power from the turbine-compressor shaft to that of the fan. The heat exchanger is a two-pass, semi-counter flow type of tubular construction. Coolant air is drawn through this aftercooler from a flush opening in the lower surface of the wing and then discharged horizontally aft from the fan through an opening in the trailing edge structure. The unit is shock mounted at the four lower corners and is quickly removable through the lower surface of the wing. A suitable section of duct will be provided to replace the unit during winter operation.

The overall dimensions of the unit are 18 in. high, 37 in. long (spanwise), and 27 in. deep. Its weight is 115 lb. The estimated increment of the supercharger drive system weight that is chargeable to the refrigeration function is 100 lb per airplane. The increase in the weight of the ducting is about 35 lb, bringing the total weight to 250 lb or 25 lb per ton of capacity for a system that ideally meets all of the airline requirements.

In order to illustrate roughly the space and weight reduction that is accomplished by use of the air turbine unit, it has been sketched in Fig. 19B to the same scale used in Fig. 19A, which shows one of the most compact commercial air conditioning units in present use. Both units have about the same capacity. The commercial unit weighs 3200 lb, the aircraft unit 115 lb. The commercial unit uses a water-cooled condenser and would be at least 50% larger if aircooled as the aircraft unit is.

The problem of ground cooling without engines operating is simplified to the extent that it requires only a supply of compressed air at about 15 psi gage pressure. This can be furnished from a central compressor in the air terminal or from individual truck-mounted units, such as shown in Fig. 20. This truck arrangement is based on using a converted aircraft turbosupercharger as the compressor by replacing the turbine with a stepup gear train. A water-cooled truck-type engine is used as the power source. A single axial fan provides coolant airflow for the truck-mounted intercooler as well as the engine radiator. A 5-kw generator supplies power for the operation of the cabin fans and other electrical equipment in the airplane.

Comparison of Installation and Performance – Fig. 21 gives a direct comparison of the significant installation and performance factors for the three systems.

It is apparent that the air cycle has a very definite advantage over the other systems when weight and space requirements are considered. From the standpoint of maximum power required it is the least favorable of the three although the duration of this maximum power load is short. As indicated in Fig. 21, the average cooling load for the Los Angeles to New York operation is only 24% of the maximum. The corresponding average power required for the air cycle is 20% of the maximum power. For the Los Angeles to San Francisco flight the corresponding average values are 71% of maximum cooling and 58% of maximum power.

The relative amounts of fuel required to operate the equipment as well as that to transport it are also shown. Of much more significance than the amount of fuel to transport the equipment is the relative sacrifice in airplane revenue which would occur in long-range operations whenever the payload is limited by the take-off gross weight of the airplane. No plot of this comparison has been shown but it would be proportional to the weight of the equipment plus operating fuel.

From the standpoint of ease of maintenance the air cycle is far superior to the other two systems for the following reasons:

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WEIGHT REDUCTION

Of Aircraft Braking Systems through Use of Reverse-Thrust Propellers

by **WENDELL E. ELDRED**
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Fort Worth Division
Consolidated Vultee Aircraft Corp.

HENRY H. KERR
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and **HAROLD H. WARDEN**
Manager, Installations Department
Propeller Division
Curtiss-Wright Corp.

THE weight savings possibilities of reverse-pitch propellers has provided one of the answers to a major problem confronting all aircraft manufacturers during the wartime years. This problem was the necessity of maintaining performance in the face of weight additions which constantly occurred as modifications and equipment were added as the result of combat experience.

By applying reverse-pitch propellers to the 100,000-lb B-32 airplane, a net weight saving of 407 lb weight empty was realized. It is interesting to note, now that peace has come, that in addition, the use of reverse-pitch propellers has a number of distinct advantages for commercial operation other than weight saving. All advantages will be amplified later in this paper, but for the present it can be said that reverse-pitch propellers also provide greater safety, greater efficiency, and greater passenger comfort in the operation of large, high-performance, land planes.

Consolidated Vultee Aircraft Corp. and the Air Technical Service Command joined efforts to confront this problem of increasing weight and decreasing performance by instituting a program of weight control for the B-32 airplane before it reached production status. It was apparent that one of the items for consideration was the brake system, which represented a high percentage of parasite weight. It was decided, in view of the somewhat limited information available, to conduct a thorough test program to determine the weight savings possible, the best operating procedures, and other pertinent information.

The Propeller Division, Curtiss-Wright Corp. and Hayes Industries, Inc., actively participated in planning and conducting those tests. While the tests were conducted for a particular airplane, it is believed that the results will

THE use of reverse-pitch propellers to assist in braking of large, high-performance landplanes, the authors report, has many advantages for commercial airline operators.

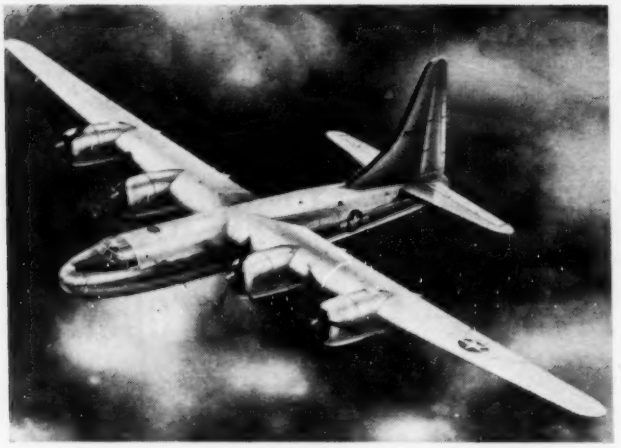
Weight savings are appreciable. For instance, they state that by applying reverse-pitch propellers to the 100,000-lb B-32 airplane, a net weight saving of 407 lb weight empty is realized.

There are other advantages, however, that the authors say can be attained, such as greater safety, efficiency, and passenger comfort.

For these advantages to be gained, the authors warn that the proper techniques of landing, using the reverse-pitch mechanism and brakes, must be used. They also believe it to be desirable that positive control surface locks be installed and that power steering be used. It appears further that steel blades should be used to withstand normal and sometimes severe abrasions.

THE AUTHORS: WENDELL E. ELDRED has been associated with Consolidated Vultee Aircraft Corp. for the past 10 years in the field of landing gear design. He is an alumnus of the University of Michigan. HENRY H. KERR (M '30), who has been an executive most of his engineering career, is now manager of aeronautical sales, Hayes Industries, Inc. A graduate from Cornell University, Mr. Kerr has been engaged successively in design and engineering sales of cutting tools with Illinois Tool Works; manager of Rapid City Air Lines; assistant manager of the Landing Gear Division, Bendix Products Corp.; and West Coast sales manager, Weatherhead Co. HAROLD H. WARDEN (M '43), manager of the installations department of the Propeller Division, Curtiss-Wright Corp., became interested in aviation while in high school and began by designing, building and flying a glider at the age of 17. He soon established himself in the field by joining American Airlines, Inc., as project engineer on flight test work, and he remained with the company until 1941, when he went to the Propeller Division.

[This paper was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Calif., Oct. 4, 1945.]



■ Fig. 1 – B-32 in flight

also prove interesting and useful to other manufacturers and operators.

I. AIRPLANE SECTION

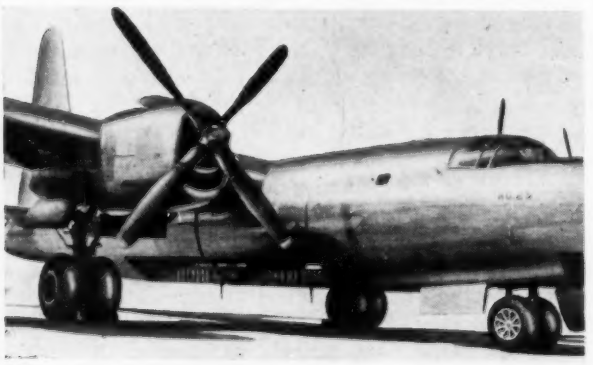
■ Test Program

Tests were conducted on a B-32 airplane (provided by the ATSC for this purpose, at Tarrant Field, Fort Worth, Tex.) by Consolidated Vultee personnel and the coauthors. They were planned in three basic phases.

The first phase consisted of making 2-engine and 4-engine reversed-propeller landing runs with and without brakes; and finally with brakes alone. This was accomplished by equipping the airplane successively with full, two-thirds, and one-half capacity brakes.

The second phase consisted of accelerated forward thrust taxi runs using brakes of the three different capacities to determine the relative taxi limitations of the brakes. Taxi runs were also made using 2-engine reverse pitch (both inboard and outboard engines) without brakes to determine the handling qualities of the airplane and the cooling characteristics of the powerplants.

Phase three consisted of static thrust tests to determine the torque capacity of the brakes in terms of engine power and brake pressure.



■ Fig. 2 – B-32 on ground

■ Test Equipment

The B-32 airplane used has a normal gross weight of 100,000 lb. The tests were conducted at an average gross weight of 92,500 lb with a wing loading of 65 psf (Fig. 1). The airplane is powered by four Wright R-3350 engines each having a take-off rating of 2200 hp at 2800 rpm. The engines are equipped with Curtiss electric 4-blade, 16 ft 8 in. diameter, reversible, full-feathering, constant-speed propellers, provided with automatic synchronization control in the forward pitch range (Fig. 2).

The standard main wheel and brake equipment was four Hayes Industries 56-in., smooth-contour, magnesium wheels with dual duplex 20 x 2¾ expander tube drum brakes. The half-brakes were attained by disconnecting the outboard brakes on this equipment. The two-thirds capacity brakes were provided by Hayes Industries specially for these tests, and were 18 x 4 dual brakes installed in 19.00 x 23 wheels. The Goodrich Tire & Rubber Co. provided the 19.00 x 23 16-ply nylon tires and tubes for this wheel size.

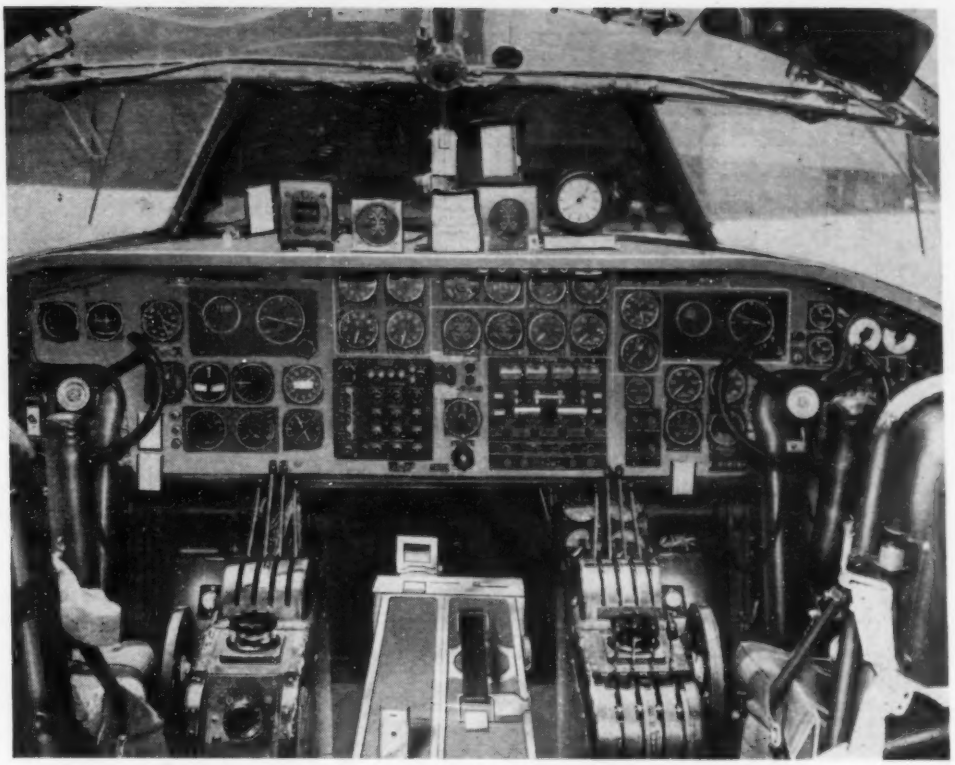
■ Instrumentation

A photorecorder was trained on the instrument panel (Fig. 3) for the purpose of recording the pertinent instruments. These registered engine rpm, manifold pressure, oil and carburetor air temperatures, and air speed. In addition, a sweep-second clock, a decelerometer, and temperature gages reading brake lining, engine turbo, and engine accessory compartment temperatures were added. Six tel-lites were also installed. One was lighted by pushing a switch at ground contact. The second was lighted when the reverse-pitch actuation button was pushed. The remaining four lighted as full reverse pitch was reached on the respective propellers. Brake pressures at the power brake valve and at the brake were recorded on two Esterline Angus automatic recorders (Fig. 4). These units provided a continuous graph recording of pressure versus time.

In order to simplify the usually difficult problem of measuring distances by means of photographic grids and other means, a switch actuated by each main gear wheel revolution was connected to a high-voltage circuit which produced a spark gap. This burned a neat hole in the paper strip of one brake pressure recorder. By counting the holes, measuring the distances between them, and knowing the tire rolling radius, it was easy to determine the velocity and distance covered at any particular time. These were plotted and gave the velocity, deceleration, and distance curves. Wind velocities were secured frequently from the control tower and also measured on the ground by means of a velometer in order to give a correction to the air-speed indicator reading to give a check on true ground contact speed.

■ Landing Test Results

A total of 70 landings and taxi runs was made with various combinations of wheel brakes and reversed propellers. Several runs were made for each condition to compensate for variations in pilot technique and instru-



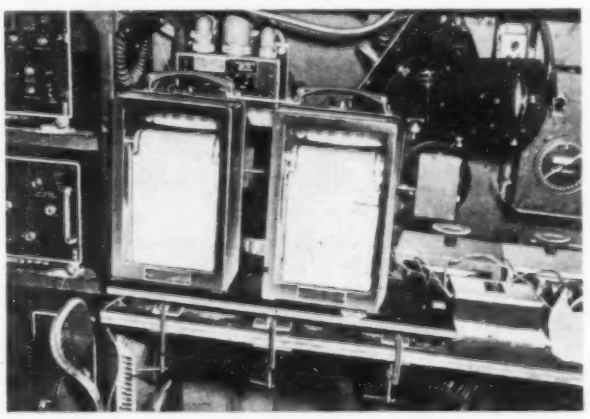
■ Fig. 3 – Instrument panel of test airplane

mentation. When the three initial runs for a given condition gave consistent results, these were used; but where any appreciable difference in results was encountered, additional runs were made to ensure reliable results.

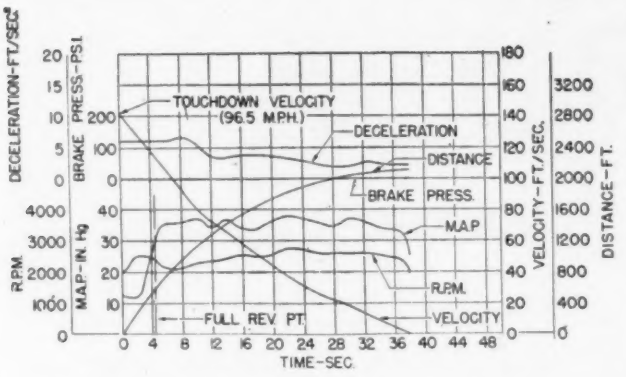
Fig. 5 is a typical landing run using no brakes and two propellers reversed. The curves are plotted with landing run times as the abscissa. Zero time represents the moment of main wheel ground contact. The distance curve shows the ground distance traveled at any interval of time following contact, and the last point on the curve is the total landing run distance, which in this case is 2109 ft. The velocity curve gives the true ground contact landing speed of 141.5 fps (96.5 mph) as its initial point and the airplane velocity at any time for the remainder of the run. The deceleration curve is derived from the velocity curve and shows airplane deceleration was a maximum of 6.0 ft per sec per sec near the beginning of the run but reduced appreciably towards the end of the run. The next two curves show the rpm and manifold pressure of the engines of the reversed propellers. The propellers were reversed at the moment of ground contact and the vertical line intersecting the rpm and manifold pressure curves show where full reversing was accomplished. This time averaged very close to 4.4 sec for all operations. Since no brakes were used, the brake pressure curve shows a zero value for the entire run.

Fig. 6 is a typical landing using full 20 x 2¾ dual duplex brakes and no reverse pitch. The brake pressure curve shows the amount of brake pressure applied during the landing run. The brake valve was set to give a maximum pressure of 200 lb. The landing distance was 2148 ft. This slightly longer stopping run was due to the higher landing speed which increased the kinetic energy 27%. On the comparative bar graphs to follow, the landing distances have been corrected to an average landing speed to give a true comparison. The maximum deceleration was 7.0 ft per sec per sec. It will be noted that the landing deceleration was more nearly the average over the whole run.

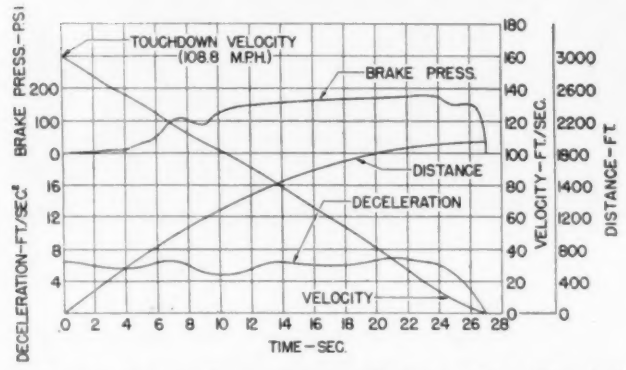
Fig. 7 gives the results of a run using four reversed propellers but no brakes. As will be noted, the landing run decreased to 1766 ft and the maximum deceleration increased to 8.5 ft per sec per sec. It may be seen that the maximum deceleration occurred 7½ sec after landing at a velocity of 100 fps (68 mph) before the manifold pressure peaked and before full engine rpm was reached at 14 sec time. This is logical when all the aerodynamic drag forces on the airplane are considered. There is appreciable aerodynamic drag of the wing, fuselage, landing gear, and other parts of the airplane. As this varies with the square



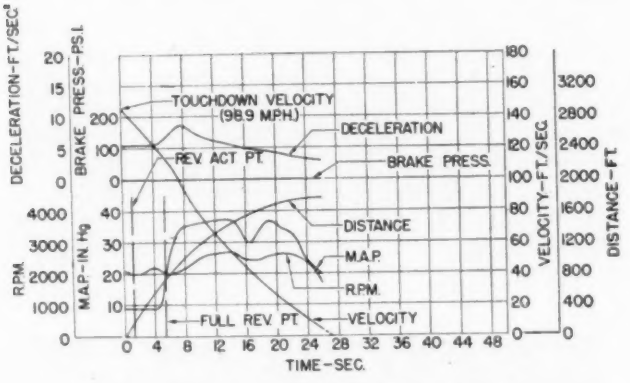
■ Fig. 4 – Brake pressure recorder installation



■ Fig. 5 – Two outboard propellers reversed – no brakes



■ Fig. 6 – No propellers reversed – full brakes (20 x 2¾ dual duplex brakes)



■ Fig. 7 – Four propellers reversed – no brakes

of the velocity, the airplane-drag-less-propeller-drag, at the velocity of peak reverse power, is only 29% of that at the velocity of maximum deceleration. Further, as shown in Fig. 8, the reverse thrust available at these two speeds has dropped about 20%. These two items tend to bring peak deceleration ahead of peak reverse-thrust powers.

Fig. 9 presents the condition employing full brakes with two engines reversed. This reduced the stopping distance to 1189 ft and raised the deceleration to 11.7 ft per sec per sec.

Fig. 10 shows the results of combining the full amount of both the aerodynamic and thermal braking systems; that is, all four propellers reversed with full wheel brakes applied. This combination gave a landing run of only 857 ft, or less than 11 times the length of the airplane. The landing run lasted only 12.5 sec. During this run the average deceleration was 11.11 ft per sec per sec and the maximum reached 17.75 ft per sec per sec. This value is about one-half *g* and is believed rather remarkable because of its magnitude and because it was accomplished with no discomfort to the occupants of the plane or use of safety belts.

Two additional conditions of interest were investigated and are shown in Fig. 11. They consisted of landing runs using no brakes, first with two engines reversed, and second with four engines reversed; and in both cases using only sufficient power to keep the engines from stopping. The 4-engine reverse low-power run was only 2570 ft long and is comparable to a moderate braked landing run. The 2-engine reverse, low-power run was 4020 ft long and is not considered excessive in that the only stopping forces were the small aerodynamic drag of the airplane plus the drag at idling power of two normal-thrust and two reverse-thrust propellers. The possibilities of low reverse power landings will be discussed later.

Fig. 12 presents a bar graph comparison of the stopping distances of the preceding brake and propeller combinations. In order to make these results as nearly comparable as possible, all the landing speeds were averaged and the distances for all runs corrected to the average landing speed and kinetic energy of 101.7 mph and 31,848,000 ft-lb. These data are all based on 100% capacity brakes. The following portion of the paper will present the comparison of the various brake sizes and a more detailed discussion of the braking picture.

II. BRAKE SECTION

■ Brake Equipment

The wheel brakes tested were of three different capacities. Comparisons of these are shown in Table 1.

The production equipment consisted of 20 x 2¾-in. dual duplex expander tube brakes containing 2765 sq in. of braking surface. For simplification, these will be designated hereafter as 100% brakes. The 18 x 4-in. dual expander tube brakes contain 1810 sq in. of braking surface and will be designated hereafter as 65.5% brakes. The 20 x 2¾-in. single duplex brakes will be designated as 50% brakes. They are identical to the 100% brakes except that there are only half as many brakes as under the 100% condition.

The energy loadings for the 100% brakes are low when compared with normal design practice for military aircraft

Table 1 – Wheel Brake Capacities

Brake Size	Area of Friction Surface, sq in.	% Area	Energy per Unit Area, ft-lb per sq in.	Energy per Unit Drum Weight, ft-lb per lb
20 x 2¾ Dual Duplex	2765	100	12,310	88,200
18 x 4 Dual	1810	65.5	16,760	141,800
20 x 2¾ Single Duplex	1382	50	24,620	176,400

Note: Energy based on 92,500 lb gross weight and 105 mph landing speed (B-32 as tested).

without propeller braking. For example, these same brakes are used on all 4-engine bombers, and under overload combat conditions, the unit kinetic energy per square inch of drum reaches 20,000 ft-lb as compared to the 12,310 ft-lb loading in these tests. At such high loadings, the service life is less than desired and high brake temperature causes the brakes to fade at the end of the landing run.

Brake Performance

Fig. 13 is a plot of deceleration against distance for maximum brake stops of the 100%, 65.5%, and 50% capacity brakes. The brake pressures on the graph are averaged over the entire length of run. As may be seen, the stopping distances and decelerations are quite similar in the case of the 100% and 65% brake. Theoretically, on the basis of the brake characteristics and pressures, the stopping distance of the 65.5% brake should have been 3.7% greater than the 100% brake, while the tests produced a 6.5% shorter run. This discrepancy is quite small and was probably caused by averaging only two runs for the 65.5% brake and using only one typical run in the case of the 100% brake. The important point is that they both gave smooth nonfading stops at unit energy loadings of 12,310 and 18,760 ft-lb per sq in. of rubbed braking surface, respectively.

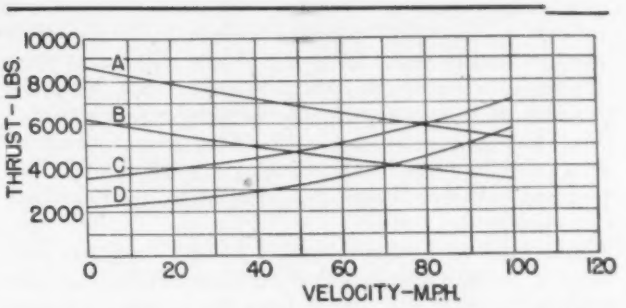
The third curve of Fig. 13 shows the stopping distance of the 50% brake to have increased to 2626 ft or about 27% more than the average of the 100% and 65.5% units. The deceleration curve shows a definite fade to about one-half the initial value due to the high energy loading of 24,620 ft-lb per sq in. This higher energy loading causes higher temperature, which in turn lowers the coefficient of friction of the lining, thus causing the fade, or lower deceleration, at the end of the run. The brake temperature at the end of the run rose from 40 C to approximately 140 C. However, after a cooling period of about 3 min, the brakes recovered sufficiently to permit taxiing around the field into position for the next take-off.

The design gross weight of the B-32 is 100,000 lb. At this weight, its landing speed is approximately 110 mph. When the 50% brakes are used, this speed and weight result in a unit energy loading of 29,400 ft-lb per sq in. of area. Laboratory tests of these brakes at the above unit loading and landing speed have been conducted. This brake made 20 stops from 110 mph at a unit heat loading of 29,400 ft-lb of kinetic energy per sq in. of area. The average stopping time was 29.1 sec; thus the average deceleration was 5.54 fps, and the stopping distance was 2340 ft. The hydraulic pressure applied to the brake to attain these stops did not exceed 220 psi.

At the end of these tests, the drum was scored and only slightly heat checked. The lining had worn 0.080 in. It was felt this equipment could make more stops; however, tests were discontinued because the brake had demonstrated its ability to stop the airplane 20 times without other braking assistance. Hence, it is concluded that the 50% brakes are thoroughly safe for several emergency airplane stops.

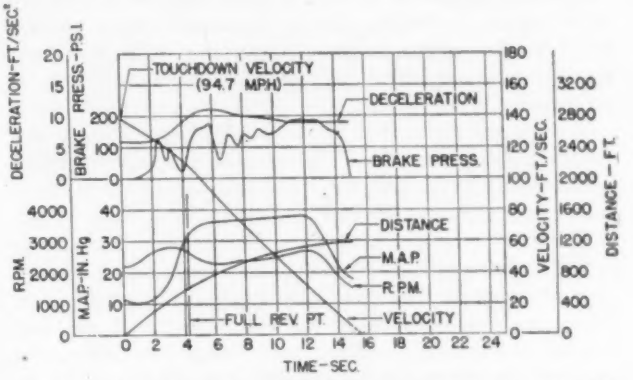
The best average deceleration obtained on any of the runs with wheel brakes only was 6.375 ft per sec per sec. This value appears low compared to the Air Technical Service Command requirements of 10 ft per sec per sec for acceptance proof test. This proof test requires a minimum average deceleration of 10 ft per sec per sec throughout a simulated landing run conducted on an inertia type of flywheel dynamometer (Fig. 14). The flywheel is so weighted that when rotating at a peripheral speed equal to the aircraft landing speed, it contains the same kinetic energy as the airplane. The tire is brought to bear against the flywheel with a load equal to the static load on the airplane and full brake pressure is immediately applied.

During all of the landing runs with brakes on, continuous effort was made to increase the average deceleration. Five tires were blown out in separate runs during these attempts. When high pressures were applied at high

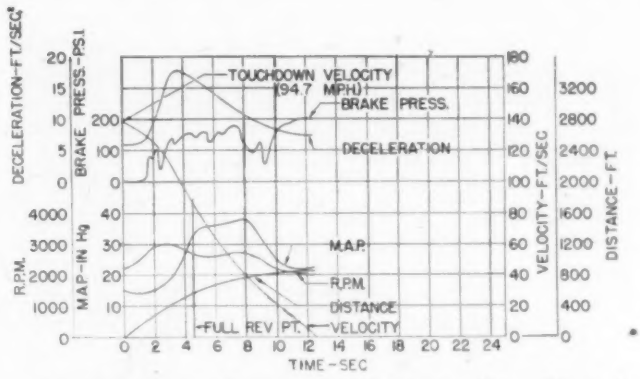


■ Fig. 8 – Comparison of forward and reverse thrusts – 4-blade propeller, 16 ft 8 in. diametor; 93.8 activity factor blade; constant-speed operation

A – Forward thrust, 2000 hp, 2400 rpm (840 propeller rpm)
B – Forward thrust, 1000 hp, 1800 rpm (630 propeller rpm)
C – Reverse thrust, 2000 hp, 2400 rpm (840 propeller rpm)
D – Reverse thrust, 1000 hp, 1800 rpm (630 propeller rpm)



■ Fig. 9 – Two outboard propellers reversed – full brakes (20 x 2¾ dual duplex brakes)



■ Fig. 10 – Four propellers reversed – full brakes (20 x 2¾ dual duplex brakes)

ground speed, tire skid resulted. At these speeds, it was difficult to detect skid in time to release the brake before the tire casing wore through. When the brakes were applied at high speed, it was necessary that the brake pressure be increased gradually to allow time for loss of wing lift, thus preventing skid. One advantage of reverse-propeller thrust braking is that it spoils the lift, thus permitting higher brake thrust at high speed without endangering the tire.

It is believed pertinent at this point to mention that none of the blowouts caused any difficulty in handling the airplane. In fact the pilot did not know that a tire had blown until the run was completed in several instances. This points to the desirability of the dual-wheel gear as compared to a single-wheel type.

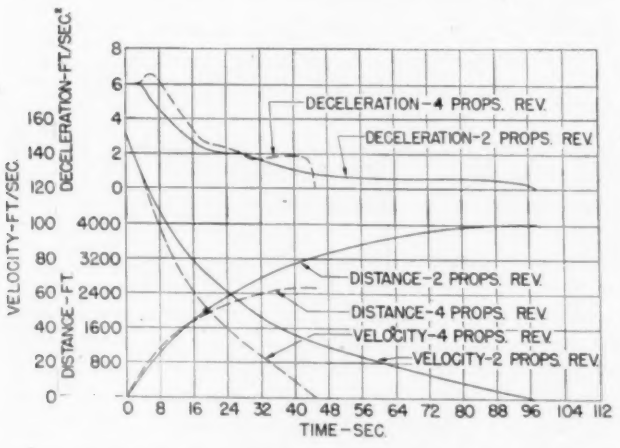
■ Optimum Operating Procedure

The effect of propeller braking only is shown in Fig. 11. These are representative landing runs made with two and four engines in reverse pitch. Minimum power was applied to the reverse engines to prevent stalling. These engines were operated at approximately 1500 rpm at 20 in. map.

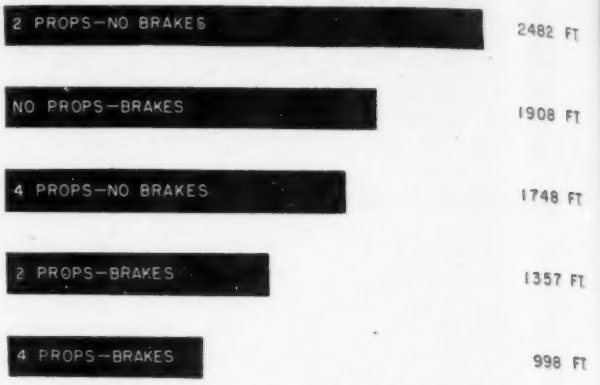
A stopping distance of 4020 ft was obtained using two reversed idling propellers. The curves show that reversible-pitch propellers produce their greatest thrust at high speed where wheel brakes are least effective.

The best operating procedure for a smooth, even stop would be to use two reversed propellers at fairly low engine powers and then gradually apply wheel brakes until moderate braking is being used at the end of the run. Assuming a desired deceleration of 5 ft per sec per sec the landing distance would be reduced to approximately 2220 ft. If this technique were followed the wheel brakes would absorb 14,280,000 ft-lb of kinetic energy and the unit energy loading would be 10,310 ft-lb per sq in. of brake area for the 50% brakes.

This energy loading is so low that these brakes would have an extremely long life and consequently low maintenance cost. This technique represents a minimum use of wheel and propeller braking and results in a satisfactory stopping distance and longer tire life.



■ Fig. 11 – Effect of two and four propellers reversed – no power and no brakes



■ Fig. 12 – Comparative stopping distance bar graph

From the foregoing data it is concluded that the 50% brakes will stop the B-32 at 100,000-lb gross weight and 110 mph landing speed. No emergency wheel brake system is required as the absence of reverse thrust from both pairs of propellers, plus the failure of the main braking system, is a more remote possibility than the failure of both normal and emergency braking system on an airplane without reverse-pitch propellers. Short emergency stops within 12 airplane lengths can be made from 100 mph with four engines reversed with power plus brakes.

■ Brake Taxi Runs

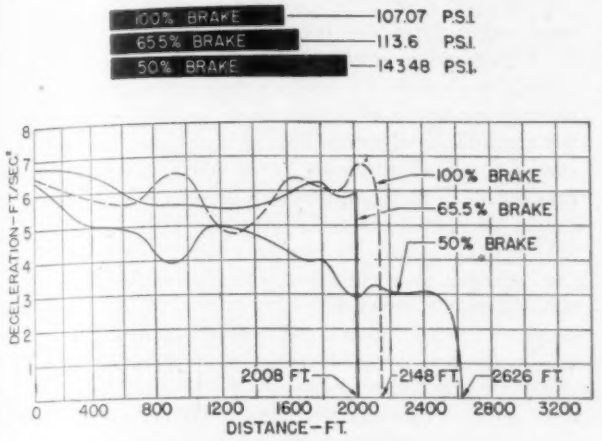
Taxi runs were conducted to determine ability of the 100%, 65.5% brakes to control speed and direction of the aircraft and to indicate the amount of energy each set of brakes would absorb before overheating.

Table 2 – Taxi Run Comparison
(Three brake conditions)

% Brake	Corrected Time to Reach Critical Temperature, % time of 100% brake
100	100
65.5	98.2
50	47

It was desired to conduct these runs in such a manner as to load the brakes beyond the requirements of normal taxi service. To this end, all four engines were operated at approximately 1500 rpm and 19 in. of mercury manifold pressure with propellers in the low pitch setting. Normal taxi procedure requires only 800 to 1000 rpm with correspondingly lower manifold pressure. Table 2 shows the relative times required to reach critical brake temperature with each set of brake conditions.

The brake manufacturer recommended that brake temperature should not exceed 143 C measured at the center of the brake block, which is 0.625 in. thick. It was found that when this temperature was attained, the brakes faded enough to make speed and directional control difficult. Therefore, 143 C was considered to be the critical temperature. The times are shown as per cent of that required for the 100% brake condition. The data taken from each run were not comparable because map's, rpm's, taxi speeds, and brake pressures varied too greatly. The data were therefore, reduced to comparable figures by deriving a



■ Fig. 13 – Comparison of decelerations and brake pressures (three brake conditions)

common energy input basis. It will be seen from this table that when reduced to comparable basis, the relative capacities of each set of brakes are fairly proportional to their area, with the exception of the 65.5% brakes which almost equalled the 100% brake. On the basis of brake drum masses, these figures are also fairly proportional.

The superior taxiing characteristics of the 65.5% brakes are explained by the fact that the brake drums of the 100% and 50% capacity brakes are entirely within the wheel outline, while about half of the 65.5% brake extended outboard of the wheel into the slipstream of the propellers. This increased the heat radiation rate of the assembly and made it a better taxiing brake design.

The purpose of this paper is to determine weight reduction resulting from the use of reverse-thrust propellers combined with wheel brakes. Therefore, the taxi run with 50% brakes is of special interest. Using 50% brakes, the airplane was taxied for 2½ min and all four engines were operated at approximately 1500 rpm and 19 in. of mercury manifold pressure. The airplane was restrained to a speed of approximately 16.4 mph by means of wheel brakes. At the end of this run, the brake temperature was 125 C and the pilot reported the brakes weak. After a cooling period of approximately 3 min, the brakes recovered sufficiently to handle taxiing to the take-off point and were used on subsequent landing runs. Under normal taxiing conditions, using 800 rpm, the taxiing time available would be about 8 min and over 2 miles in distance.

It is concluded from these tests that 50% brakes are adequate under normal taxi conditions. It will later be shown that taxiing with two reversed propellers and two in forward-thrust position is entirely practicable; therefore, the use of wheel brakes alone for taxiing should be considered an emergency condition.

■ Static Thrust Tests

Static thrust tests were conducted to determine the ability of each of the brake combinations to hold two and four engines at 2200 hp, 2800 rpm, and 49 in. of mercury manifold pressure. The 100% and 65.5% brakes held four engines with brake pressure below 200 psi. The 50% brakes would not hold all four engines at 205 psi, which was the maximum brake pressure the hydraulic system would permit. However, the 50% brakes held two engines at full power with 178 psi line pressure. If engines are tested at full take-off power in pairs, as is normally done, the 50% brakes will adequately prevent the airplane from creeping.

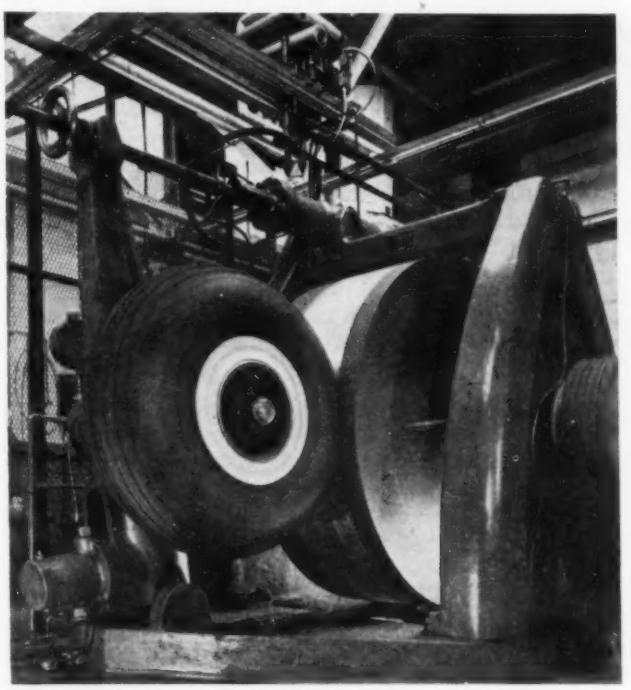
■ Condition of Equipment After Tests

Four of the eight 20 x 2¾-in. duplex brakes were operated as part of the 100% brake condition for 25 landing runs and one taxi test similar to that described for the 50% brakes, and then were continued as the 50% brakes. Under this condition, they made 11 landing runs, two of which were emergency brake stops. In addition, they accomplished the taxi run previously described.

Examination of these four brakes and their drums at the conclusion of this test showed the drums to be in excellent condition. The frictional surfaces of the drums were smooth and polished. Small heat checks were beginning to appear on the surface. Centrifuge brake drums can be operated until the heat checks progress completely through the cast-iron liners; therefore, these same drums could have been continued in regular aircraft service for many more landings.

The brake lining was quite badly deteriorated. Approximately 10% of the lining surface was gouged to a depth of 1/16 in. These gouges were of large area, indicating that the lining had been disintegrated by heat. Therefore, the linings had, in making all of these stops, performed their full service.

These observations indicate that the lining is the critical part. Hence, a brake to withstand 100% overloads even for a small number of landings should contain the maximum amount of lining, for example, the lining should cover approximately 100% of the total braking surface area. Also, the pressure loading throughout the braking



■ Fig. 14 – Inertia brake dynamometer

surface should be uniform, otherwise the metallic friction member will suffer distortions from uneven pressures, causing warping and more complete heat checking with consequent rapid failure of this member.

■ Weight Saving

Fig. 15 shows the comparative weight savings achieved on the B-32 by use of 65.5% brakes and 50% combined with reversible-pitch propellers. This bar graph shows brake, propeller, and net weight savings compared with the 100% brake condition. The weights of the hydraulic system are not included, but the savings amount to 20 and 50 lb for the 65.5 and 50% brakes, respectively.

For the 100% and 65.5% brakes, dual-brake wheels are used; however, for the 50% brakes, single-brake wheels are desirable. This reduces complexity, expedites tire change, and eliminates the splined axle extension and outboard brake flange. Thus, in addition to a weight saving, with its consequent increased payload or range, there will be a considerable saving in cost of initial equipment.

Surely 407 lb increased payload (or increased range of 64 miles), decreased maintenance, greater simplicity, and lower initial cost warrant the use of reversible-pitch propellers. Therefore, the characteristics, operational technique, and other advantages of this propeller feature will be discussed in the following section.

III. PROPELLER SECTION

■ Propeller Equipment

The four reversible propellers used in these tests, described previously, were each equipped with four hollow steel blades of production design, each blade having an activity factor of 93.8 and a thickness ratio of 0.082. The maximum negative blade angle was set at −15.5 deg, which permitted a reverse thrust of approximately 6000 lb per propeller at 120 mph, using 75% normal rated power and 91% normal rated engine speed in the static condition.

Comparison between forward- and reverse-thrust variations with air speed, is shown in Fig. 8 for two power conditions. As shown, the reverse thrust available at 100 mph using 2000 hp at 2400 rpm is approximately 83% of static forward thrust and decreases to approximately 41% of static forward thrust at 0 mph.

For comparison of constant-speed, reverse-propeller operation versus fixed-pitch reverse-propeller operation, see Sheets and MacKinney.[1]

Referring to our three primary objectives, the first is now considered.

■ Safety

Safety is increased by the presence of two distinctly different braking systems. Although the wheel brake system size can be greatly reduced when used with propeller braking, it can serve alone, in an emergency, to stop the aircraft within the specified limit as proved by these tests. (See Fig. 16.) Further, much shorter landings are possible when both wheel and propeller brakes are used simultaneously, since they supplement each other in a perfect manner. (See Fig. 17.) This braking ability cannot be over-stressed due to the possible saving of life and equipment when forced to land on small emergency airports or if obstructions accidentally appear on runways of adequate size airports. Also, since propeller braking is equally effective on icy or wet runways as on dry concrete, an important additional safety feature is realized. Several cases have been reported giving reverse-thrust credit for saving production B-29 airplanes from destruction, and possibly saving their crews as well.

Due to the ability to reverse either the two inboard or the two outboard propellers, as well as all four simultaneously, a major safety feature exists. In the event of an engine failure, the remaining pair of unaffected propellers can be reversed. Fig. 18 indicates that reversed outboard propellers stop the airplane in about 15% shorter distance than reversed inboard propellers. This indicates that the reverse-pitch propellers on the inboard engines have greater propeller interference due to fuselage effect and the larger wing section at the inboard nacelles, thus reducing propeller efficiency.

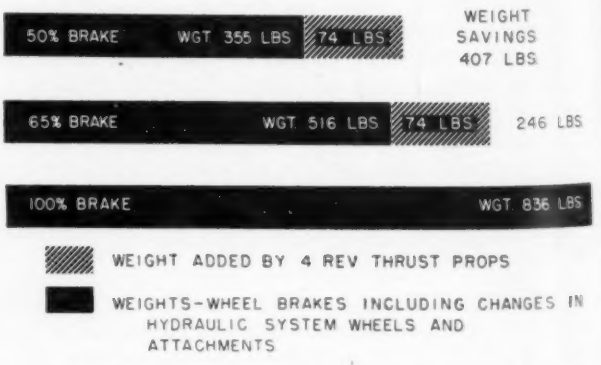
High-speed taxi tests to date indicate that three operative propellers can be reversed simultaneously to provide greater braking effect without serious difficulty. However, the use of a steerable nose wheel is believed necessary to provide satisfactory directional control. It is possible that future experience will show full control can be attained by proportional manipulation of the throttles controlling the reversed propellers.

■ Efficiency

Reduction in weight, maintenance, and cost is "efficiency" to any aircraft engineer if performance is maintained or improved. Referring to Fig. 15, the two brake systems are compared as to installation weight. Previous data showed that as much as 407 lb saving can be made.

An interesting realization of this weight saving can be had from the following: One airline estimates a saving of $30 per lb saved per airplane per year. For a relatively small fleet of 20 airplanes, the maximum weight reduction shown here would mean an annual monetary saving of $244,200. This is a very impressive figure realized from weight reduction alone.

Weights chargeable to the installation of reverse thrust on the B-32 test airplane were as follows:



■ Fig. 15 – Comparative brake weights (wheel and propeller)

[1] See SAE Transactions, Vol. 53, May, 1945, pp. 257-263: "Reverse-Thrust Propellers as Landing Brakes," by J. H. Sheets and G. W. MacKinney.

For two-propeller reversing:

22.8 lb propeller equipment
7.2 lb wiring
30.0 lb total

For four-propeller reversing:

60.6 lb propeller equipment
13.4 lb wiring
74.0 lb total

For present production airplanes the weights are reduced below these values due to more recent design. Total weight for two-propeller reversing is approximately 25.5 lb and total weight for four-propeller reversing is approximately 55 lb, thus providing an additional airplane weight saving of 20 lb for a four reversible propeller installation; a total weight reduction of 427 lb. Based on these weights, the operating cost saved per year on a fleet of 20 airplanes is $256,200 or further saving of $12,000 per year.

Saving in initial cost, when comparing the two braking systems, is considered to be quite negligible. However, due to the absence of wear from aerodynamic braking compared to wheel brake wear, it is believed that the "braking" cost throughout the life of an airplane will be markedly reduced. It is beyond the scope of this paper to prove the matter of cost and maintenance; however, present indications point to a considerable cost reduction when aerodynamic braking is used, due to less usage of wheel brakes, consequently less wear and maintenance.

In the interest of longer service life and less maintenance it is emphasized that the propeller blades should be of a rugged design and constructed of a hard alloy steel to withstand continuous reverse-thrust landing usage. Figs. 19 and 20 indicate the basis for this recommendation. Although the runways used were dry concrete, a certain amount of abrasion was experienced on the camber side of the reversed propeller blades, due to the loose foreign particles on the runway surface. Fig. 19 shows the camber face of an inboard propeller blade which is somewhat more affected by abrasion than that shown on Fig. 20 which is an outboard propeller. The difference in degree of abrasion is due primarily to the main landing gear wheels, which tend to loosen and kick into the air, runway surface sand and small pebbles. The pocked appearance is due to the absence of paint but the steel itself remained unaffected. Approximately 70 hr were on these propellers at the time the photos were made. It is believed that continuous use of aluminum-alloy blades, under similar or more severe conditions, would require frequent grinding and polishing operations which would drastically reduce the service life. In view of the above, and from limited experience to date, it is believed that steel blade service life will be relatively unaffected by reverse-thrust application.

Equipping standard propellers with the reversible feature in no way decreases the original aircraft performance efficiency, but rather improves the performance through weight reduction.

■ Passenger Comfort

Smooth deceleration is an important factor in stopping large aircraft within a limited distance, thus providing greater passenger comfort and improved controllability. Referring to Fig. 21, four typical landing deceleration curves plotted against distance are shown. The first is the result of wheel brakes only; the second is the result of four reversed propellers, power applied but no brakes used; the third is the result of four propellers reversed, no power and no brakes; and the fourth is the result of two propellers reversed, no power and no brakes.

The third and fourth curves are unique in that smooth, effective deceleration is shown, yet no reversed-thrust engine power, as such, was applied. Approximately 12% rated power was used giving 1500 rpm at 20 in. map to prevent the air from windmilling the reversed propellers backward. Propellers in the normal position were idling. The procedure of reversing the propellers but not applying power or brakes may be favored by operators as normal operating procedure, for the following reasons:

1. No high-speed engine noise produced during the landing roll.
2. No braking wear on the wheel brakes, engines, or propellers.
3. Pilot operating technique reduced to a minimum.

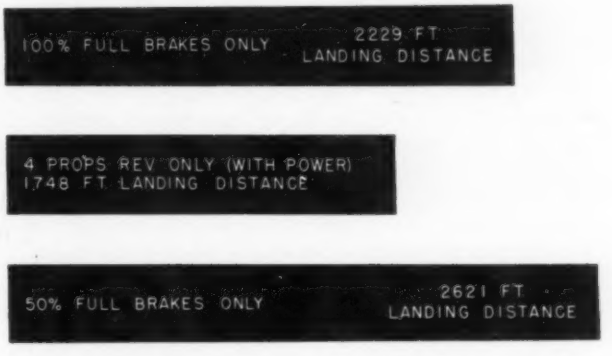
It is conceivable that an airline could make routine landings using two inboard propellers reversed, no power and no brakes. At the end of the landing roll all propellers would be left in this status and taxi to ramp would be accomplished by propeller-thrust directional control, thus saving brake wear during taxiing as well as providing smoother taxi operation. Another interesting point of comparison can be made by noting the maximum deceleration obtained from previously discussed tests.

Maximum wheel brakes only *8.1* ft per sec per sec
Four propellers reversed only *10.5* ft per sec per sec (with power)

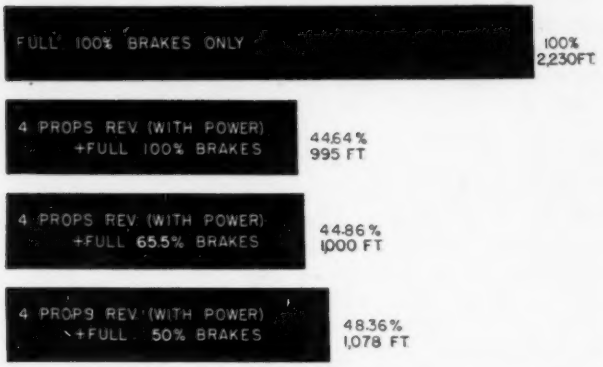
This is due to the fact that propeller braking is much more effective at high landing velocities than wheel brakes.

Referring to Fig. 21 it can be seen how great reverse thrust is at higher landing velocities. Remembering that resulting drag of reversed propellers alone without power is maximum at high velocities, the greater deceleration value of 10.5 ft per sec per sec from aerodynamic braking is fully understood.

Brake jerking has been a common pilot and passenger complaint in the past due principally to maladjusted



■ Fig. 16 – Comparison of brake effectiveness of 50% brakes and reversed propellers with 100% brakes



■ Fig. 17 – Comparison of four propellers reversed plus 100%, 65.5%, and 50% brakes with 100% brakes only

hydraulic system, unequal pressure applied by the pilot, brakes of unequal wear, and the like. Realizing these factors disappear when aerodynamic braking is used, it is expected that future experience will also reveal a smoother and more uniform landing-roll stop.

From data presented earlier it is obvious that operating technique during these test runs had marked effect on the reduction of landing roll and passenger comfort. With regard to propeller-reversing operating technique, recent intensive research has done much to simplify the cockpit controls and to provide a more automatic application, thereby removing some of the human element, which should tend to make each landing smoother and more uniform with consistent brake and/or power application. The present results of this research were not available at the time of these tests.

It now appears that a reverse-sense throttle, designed to actuate the reversing cycle, provide the necessary time delay, and apply power in the proper sense will be a valuable contribution to obtaining the desired operational functions. Several experimental installations are now utilizing the reverse-sense throttle. The B-32 test installation used standard forward-sense throttles for power application when the propellers were in the reversed position as well as when in the normal forward position. The reversing circuit for either the outboards, inboards, or all four propellers was selected by actuating either or both of two reverse selector switches mounted on the instrument panel, accessible to the copilot. When the reversing cycle actuation was desired, either of the momentary push-button switches located at both the pilot and copilot stations, were pushed subsequent to the setting of the selector switch. A delay period of approximately two seconds was held between this operation and the opening of the throttles of the reversed propellers. The full-reverse pitch angle of −15.5 deg was reached on an average of 4.4 sec from the time the reversing cycle was actuated. (See Figs. 9 or 10.) Past experience indicates that the average pilot can gain adequate familiarization with this procedure after making three or four reverse-pitch landings. It is very probable that reverse-sense throttles will lessen this familiarization time.

Jesse Reese, Consolidated Vultee test pilot on these tests, reported considerable control surface buffeting and some directional control instability during four-propeller, reverse-thrust landings using full power. It was his conclusion, however, that the use of positive control surface locks and a steerable nose wheel would satisfactorily correct these faults, thus greatly improving passenger comfort. The locks should be designed to engage in any position of the surface controls and preferably be operated by a single control. Further, due to rudder buffeting, constant brake pressure was difficult to maintain. Locked controls would correct this difficulty and permit improved wheel brake effectiveness.

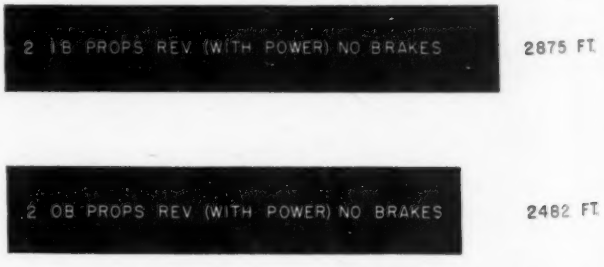
■ Propeller Taxi Tests

Several of the taxi runs made were particularly significant as to reversed-propeller operation. A run was made using reversed inboard propellers, normal outboard propellers, and no brakes. A fairly low taxi speed averaging 14.25 mph (with a maximum taxi speed of 19.5 mph and a minimum taxi speed of 8.76 mph) was maintained purposely to determine engine cooling characteristics for a distance of 2.85 miles and a period of 12 min. At the 12-min duration point the engine oil temperature of a reversed-propeller engine reached its limit of 110 C from a starting oil temperature of 90 C. The outside air temperature was 30 C, rpm and map of the inboard engines was held quite constant at approximately 900-1000 and 19-20, respectively, except at turns of the triangular course where changes were made to permit steering by propeller thrust alone. Rpm of the outboard engines varied between 18 and 20 in. of mercury. Twelve-minute taxi time far exceeds that normally required on the average airport.

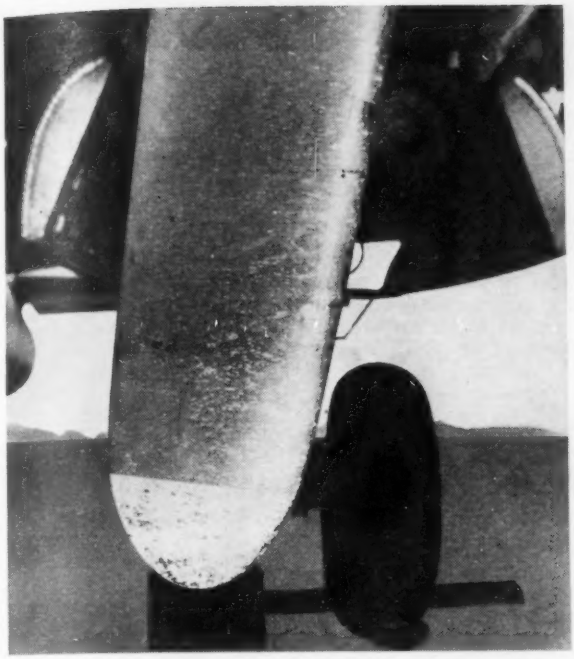
During this run, engine accessory compartment temperatures were constantly checked at the carburetor and the turbo wastegate locations; however, no noticeable rise in temperature occurred. Cowl flaps were full open during all taxi and landing operations.

Steerability was easily accomplished by throttle manipulation alone, no brakes being used, even to the point of taxiing the airplane through narrow passageways and backing into the final parking place. It was apparent, however, that the reverse-sense throttles should be an aid in simplifying taxiing by reverse thrust only.

A somewhat similar taxi run was repeated using the outboard propellers reversed and the inboard propellers normal, the purpose being to determine engine cooling variation between the inboard and outboard engines operating with reversed propellers. Rpm and map were held consistent with those of the previous run, and steering



■ Fig. 18 – Comparison of reverse-thrust effectiveness of two inboard propellers versus two outboard propellers



■ Fig. 19 – Reverse-thrust effect on inboard propeller blade

was again accomplished by propeller thrust alone, no brakes being used.

A slightly slower taxi speed was used to determine the maximum time the engines could operate with little or no air available for cooling. An average taxi speed of 10.08 mph was maintained for a distance of 2.52 miles (with a maximum speed of 19.2 mph and a minimum speed of 8.3 mph) for 15 min. At the end of this time no temperatures had reached their limits. It is concluded that a continuous taxi time of 15 min, under the above conditions and with an outside air temperature of 30 C, is sufficient to show the practicability of this type of operation. From the foregoing it is indicated that with propellers reversed the outboard engines on the B-32 operate slightly cooler than the inboard engines probably due to: combination of power and rpm used, difference in wing-nacelle configuration, and body interference effect on airflow; also that taxiing of aircraft by propeller thrust alone is practical from an engine cooling point of view.

■ General Conclusions

The following general conclusions are those of the three authors based on the foregoing tests and data:

1. Approximately 407 lb (47.8% of wheel brake system weight) can be saved by use of four reversible propellers on a 92,500-lb B-32 type airplane and elimination of one-half the wheel brake capacity.

2. Brake dynamometer tests have conservatively shown that brakes of 50% capacity have emergency capacity to stop the B-32 airplane at 100,000 lb gross weight in excess of 20 landings without the use of reverse-pitch propellers.

3. It is, therefore, concluded from (2) above, assuming equivalent brake and power loadings, that brake capacities may be cut in half and that emergency brake systems may be eliminated on both 2- and 4-engine airplanes by the use of reverse-pitch propellers.

4. To obtain the shortest run possible, it is necessary that the reverse-pitch mechanism be actuated immediately upon ground contact and that full power be applied as soon as possible and held for the duration of the run. Brakes should be applied as rapidly as possible without skidding the tires.

5. Use of reverse-thrust propellers permits application of greater wheel brake torques at high initial landing speed without skidding the tires, due to reduced wing lift.

6. For average landings, where only reasonable stopping distances are desired, it appears best to use two reversed-pitch propellers with only moderate power, and, as the speed begins to drop, to apply brakes gradually until moderate braking is being used at the end of the run. This will give a smooth, quiet, evenly decelerated landing with a minimum of wear on the braking equipment.

7. Reducing landing roll by reversing two or four propellers, and not applying power or brakes, appears to be an excellent normal operating procedure where the runway lengths permit.

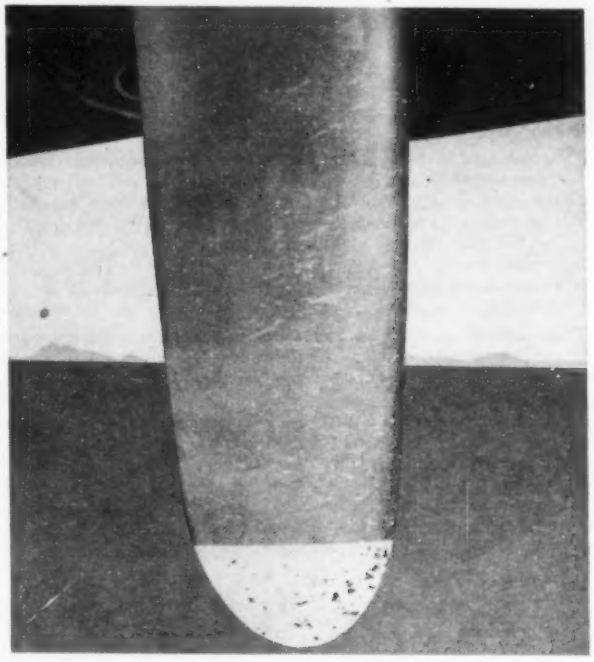
8. In order to eliminate control surface buffeting induced by reverse-thrust operation with high power, it is very desirable that positive control surface locks be installed. These locks must be able to engage at any position and preferably be operated by a single control.

9. Taxiing without brakes, using normal and reverse propeller thrust alone, is adequate, but power steering is desirable on large airplanes to simplify pilot technique. Reverse-sense throttles should also prove worth while.

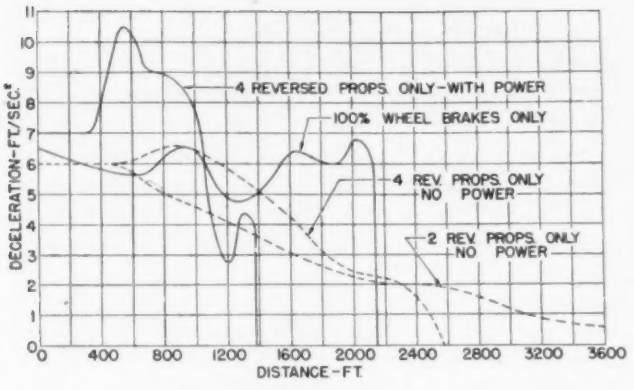
10. Engine cooling was satisfactory with either inboard or outboard propellers reversed for taxi periods less than 12 min endurance, with an outside air temperature of 85 F.

11. Engines operating reversed propellers show no tendency to overheat in normal reverse-thrust landings.

12. Safety of the aircraft is greatly improved by the



■ Fig. 20 – Reverse-thrust effect on outboard propeller blade



■ Fig. 21 – Comparison of deceleration curves – wheel brakes versus reverse thrust

installation of reversible propellers supplemented by an optimum amount of wheel brakes.

13. Steel blades should be used for reverse-thrust applications to withstand normal and sometimes severe abrasion.

DISCUSSION

Care Needed in Application Of Power with R-P Braking

Excerpts from discussion of

—M. G. (Dan) Beard

American Airlines, Inc.

THE authors have recognized the fact that reverse-pitch propeller operation with high-power application creates severe turbulence conditions on the control surfaces. I first experienced this on one model of airplane where the application of rated power in the reverse pitch made it possible almost to take the controls away from the first officer. Several landings using less power indicated that a normal amount of braking can be obtained using all four propellers in reverse pitch and applying only a moderate amount of power. This procedure did not seriously buffet the control surfaces and the cockpit controls could be steadied easily by the first officer.

The authors have suggested a control lock for securing the controls when reverse pitch is applied. It should be pointed out that control locks are a matter of grave concern to all flight personnel, since several fatal accidents have occurred by controls being locked during the take-off. At the present time I believe the operators would rather accept a moderate amount of power application in reverse pitch, with only slight effect on the surface controls in place of easily applied control locks.

Two systems of operation of reverse-pitch controls are discussed by the authors; namely, opening the throttles in the normal direction for reverse-pitch power application, and retarding the throttles to the rear of the normal idling position of the engine for power application in reverse pitch. The simplest mechanism is probably a straightforward application of power at all times when power is needed. I believe the matter is largely one of training and repetition until the correct response becomes instinctive. It does appear that if reverse pitch is not used frequently enough to become a normal operation, the proponents of the reverse throttle have a good argument outweighing the one of simplicity of mechanism for the forward opening throttle.

The authors have pointed out the increased braking produced at high speeds and the value of applying reverse pitch as soon as the airplane is on the ground. Under normal operation on runways of suitable length for transport operations, there is no necessity for hasty action when operating reverse-pitch braking. In fact, the ability to perform the reverse-pitch operation without haste and without split-second timing is one feature that has sold the airline pilots and operators on the possibilities of using this type of braking in normal operation.

The necessity of applying sufficient power prior to or simultaneously with the actuation of the propeller in reverse pitch so that the engine will not stall as the propeller shifts over to reverse windmilling, which tends to stop the engine, is also touched upon by the authors. Especially in cold-weather operation, engines have too much tendency to die on landing with throttle in idling position. Add to this the braking effect in reverse pitch before the pilot applies power for reverse braking and there will be altogether too many engines that will stop on landing as the propeller is reversed and before power is applied for braking. Therefore, the mechanism that actuates the propeller to reverse pitch must also apply sufficient power to keep the engine running at some adequate speed that will prevent killing of the engine before braking power is applied.

Suggests Way to Determine Amount of Braking Required

—E. K. Laswell

Air Technical Service Command

THE instrumentation of the airplane for this project is considered excellent. Although the method of recording wheel revolutions from which distance and velocity were computed is justified on the basis of simplification, the method leads to some error because of the constantly changing rolling radius of the tire with velocity, which means that the effective tire rolling radius cannot be determined at any instant. The magnitude or direction of this error cannot be estimated as no rolling radius was mentioned in the report.

From a purely analytical standpoint it is regretted that decelerations were not obtained with all engines slow idle, propellers forward and feathered, using no brakes. It seems obvious after viewing Fig. 11 and noting the decided braking effect of reversed propellers with idle power that when engines are idled with propellers in the forward direction the wheel brakes are required to do considerably more work than would be the case with engines stopped. This is a point to be borne in mind by any pilot faced with having to make an emergency stop.

The report does not mention the wing flap position, but it is assumed that the flaps were left in the landing position. Wheel braking can be applied more effectively at higher speeds if the flaps are retracted. This results in a shorter distance of stopping, assuming that the wheel brakes have adequate capacity. This method of stopping also results in greater wear of the friction surfaces than when aerodynamic braking is employed in conjunction with the wheel brakes.

Since the data presented make some obvious comparisons between stopping distances obtained with wheel brakes only and reverse-pitch propellers alone, it is believed to be in order to point out some other conclusions along this same line which can be drawn from the data presented and to add some data taken from other airplanes using reverse-pitch propellers.

Fig. 6 is cited as a typical curve, from which it will be noted that the wheel brakes stopped the airplane from 108.8 mph in 1950 ft, yet Fig. 12 states that 1908 ft (42 ft less) would be the stopping distance from 101.7 mph. Referring back to Fig. 6, using the same rate of deceleration, and dropping down the velocity curve from 108.8 mph to 101.7 mph, then vertically downward to the distance curve a point is obtained which, if used as o ordinate and abscissa, a new stopping time of 25.4 sec and a new stopping distance of 1750 ft is obtained.

Referring to Fig. 7 it is seen that four reversed propellers stopped the airplane in 1760 ft. If the touch-down velocity had been 101.7 mph, the stopping distance would have increased to 1812 ft.

The conclusion drawn from this bit of play with curves and figures is that Fig. 12 on bar graph for "No Props – Brakes" should show 1750 ft in lieu of 1908 ft and that bar graph "4 Props – No Brakes" should show 1812 ft instead of 1748 ft.

Dropping back now to Fig. 21 it will be noted that the curve for "4 Reversed Propellers Only – With Power" drops vertically from a deceleration of 4 to o without increase of roll distance. This performance is unlike the action of reversed propellers. There should be a gradual decrease in deceleration with an increase of roll distance. Using the authors' value from Fig. 7 the distance should be 1760 ft instead of 1380 ft. The distance of roll for "100% Wheel Brakes Only" should be 1940 ft instead of 2150 ft (taken from Fig. 6).

This readjustment still makes no correction for velocity, the propellers stopped the airplane from 98.9 mph, while the wheel brakes stopped it from 108.8 mph.

Making reference once more to Fig. 21, if the curves for "4 Reversed Props Only – With Power" and "100% Wheel Brakes Only" were truly comparable, the work done in bringing the airplane to a stop in each case would have to be equal. If the work was equal the area under each curve would be the same. Since the area under the curve for "100% Wheel Brakes Only" is 28% greater than under the curve for "4 Reversed Props Only – With Power," the wheel brakes obviously had more work to do.

If the stopping distances computed above for adjusted velocity of 101.7 mph, of 1750 ft for wheel brakes and 1812 ft for reversed propellers, the area under the curves should be more nearly the same and the curves consequently more comparable.

The unevenness of the deceleration curve could be attributed to poor controllability, and the failure of the brakes to produce a deceleration higher than 6.8 to insufficient brake pressure. Data are available to show that brakes identical to those used on the B-32 airplane stopped another airplane weighing 130,000 lb (B-32 weight 92,500 lb) from 119 mph in 1800 ft and produced a deceleration at times of over 10 ft per sec per sec. The same brakes also stopped an airplane weighing 138,000 lb from 126 mph in 2800 ft. Using identical engines and propellers to those used on the B-32 airplane the reversed propellers required 3500 ft to stop the 138,000-lb airplane from 124 mph. This airplane had a very controllable, low lag, brake control system, and it was necessary to use a maximum pressure of 325 psi on the brakes.

On another four-engine airplane weighing 62,500 lb, the brakes stopped the airplane from 94 mph in 1500 ft. Four reversed propellers, full power, stopped this airplane in 1770 ft from 96 mph. Fifty per cent brakes only made the stop in 1750 ft from 97 mph.

Obviously, the brakes on the B-32 airplane were not being used to anywhere near their capacity. As a matter of fact the brakes were designed for the B-32 at 120,000 lb and for another type of airplane at 140,000 lb in emergency. Under wartime service conditions the B-32 would have found itself loaded to at least 130,000 lb, under which conditions its 100% brakes would have been needed, or reversible-pitch propellers combined with some percentage of the 100% brakes.

In order to determine the amount of brake required a number of "musts" will have to be established. The most important of these are considered to be as follows:

1. Hold normal run up of engines.

2. In combination with one-half the total number of reversed propellers on the airplane, produce an average deceleration of 10 ft per sec per sec, as a normal service working condition. (Single-engine airplanes should have brakes for full energy absorption with no allowance for reversible-pitch propellers.)

3. By being over-pressured, produce decelerations of 10 ft per sec per sec for short intervals of time.

4. Stop the airplane without aid of propellers at average rate of deceleration of 6 ft per sec per sec.

All these things the brakes must do for the airplane at its maximum gross weight and take-off speed. If there is one time above all others that airplane decelerating devices must function quickly and positively it is in the event of the plane's refusal of take-off after reaching near take-off speed.

Commenting on the general conclusions of the authors, in order of their listing:

1. The amount of weight saving cannot be concurred in, for had the brakes been designed originally for a 92,500-lb airplane instead of 120,000 lb, neither their capacity nor weight would have been nearly as great. It would, therefore, have been impossible to reduce the weight of the brake system by 47.8%. Twenty per cent is believed to be a more nearly usable figure for design practice.

3. For the reason given in the above paragraph, it is not believed possible to cut wheel brake capacity in half. It is agreed that reverse-pitch propellers will allow the elimination of separate emergency wheel brake systems.

8. The need for a control lock is concurred in, but the design and location of this lock must be the subject of intense and careful study by the airplane manufacturers and operators. It must be realized that the addition of reverse-pitch propeller and control locks introduces at least two more potential accident makers, and that the loss of one airplane, considering nothing but the dollars, would pay for a large amount of excess wheel brake weight and maintenance.

9. It cannot be agreed that taxiing with combinations of engines and propellers alone is adequate, particularly on busy airports. The operation is too slow. Power steering is considered mandatory and even then some wheel brake would have to be used for snappy operation about airports.

13. From observation very little loose material is thrown up by the tires as stated by the authors, but the reversed-pitch propellers swept a large area of the runway and threw dust and particles ahead of the wing where the dirt reversed direction and swirled with violence. It seems possible that the propellers may in this manner come in contact with some abrasive particles several times in succession.

It is urged that no one regard the above discussion as purely argumentative or in any way derogatory. It was felt that this paper painted too rosy a picture for reverse-pitch propellers and also presented comparisons between wheel brakes and reverse-pitch propellers which were not borne out by tests on other airplanes, although the authors had no way of knowing this.

Refrigeration for Transport Aircraft

continued from page 106

1. The equipment is small and light and contained in a single, integral unit.

2. It can be located in an easily accessible part of the airplane which is remote from the cabin and incidentally releases valuable cargo space consumed by either of the other systems.

3. Because the air itself is the refrigerant, small amounts of leakage are tolerable in this system, while in the Freon or alcohol systems any leakage is serious.

4. No heat exchanger is used to cool the air, thus reducing the pressure-drop requirements of the cabin fans and eliminating the need for cleaning the coils that are used in both of the other systems.

5. Except for the coolant fan and its driving gear, this unit could be considered a one-moving-part mechanism.

6. The cycle is so simple that no highly specialized training program will be required for the mechanics who will service this equipment.

■ Conclusions

The air turbine refrigeration cycle has been shown to be the simplest, lightest, and most compact system for the purpose of air conditioning pressurized transport airplanes. This is due at least partly to the fact that the primary compressors are included as elements of the cabin supercharging equipment.

If the weight estimates of the Freon system are valid, there is reason to believe that even for an unpressurized airplane the overall weight of the air turbine system, including the entire burden of the primary compressors, would be slightly less than that of the Freon system.

■ Acknowledgment

The author wishes to acknowledge the assistance of H. M. Cousins of the Thermodynamics Group of the Lockheed Engineering Department. Mr. Cousins contributed valuable help in the preparation of design calculations upon which many of the conclusions of the paper are based.

SILICONES as L

THE last three or four years have witnessed a remarkable development in the chemistry and general technology of silicone polymers. The first silicone polymer to be made available was a resin which found its principal application as an electrical insulating varnish. At the present time the family of silicone polymers has expanded to include materials as far apart in properties as gasoline is from rubber. In general the silicones can be classified in four groups according to their physical properties:

I. Varnishes and Resins: The resinous silicones, when used to insulate electric motors, generators, and other equipment, offer several outstanding advantages because of their stability to heat and resistance to moisture:

(a) Greatly prolonged life of equipment operating under adverse service conditions.

(b) Greater freedom from overload failures and reduced fire hazards.

(c) Increased horsepower output per unit weight accomplished through redesigning or rewinding with silicone insulation.

Silicone resins are also used in formulating protective coatings highly resistant to heat, excessive moisture, oxidation, and ultraviolet rays. Silicone-base paints and enamels which withstand temperatures up to 500 F are used for coating ovens, smoke stacks, and exhaust equipment of internal-combustion engines.

II. Silicone Rubbers: This is a group of rubbery, organosilicon oxide polymers characterized by exceptional heat resistance and retention of flexibility at temperatures as low as —70 F. Silicone rubbers can be obtained in molded or extruded shapes, and are also used to coat glass fabrics for high-temperature gasket and diaphragm applications.

III. Greases and Compounds: Silicone electrical compounds, greaselike materials which change very little in consistency over the temperature range of —40 F to 400 F, have been used widely as auxiliary dielectric materials in ignition systems of aircraft and in disconnect junctions of radio systems. Other silicone greases have been developed for high- and low-temperature lubrication uses, and these will be discussed in more detail later.

IV. Silicone Fluids: This is a group of liquid polymers obtainable in a very wide viscosity range and includes volatile fluids as well as nonvolatile oils. The silicone fluids are heat stable, inert, have good electrical properties, and are insoluble in water. Table 1 lists the silicone fluids available at present and shows a tabulation of their physical properties.

The purpose of this paper is to discuss the relationship of the general properties of silicone fluids to lubrication applications, and to describe some of the experiences to date with the lubricating behavior of silicones.

[This paper was presented at the SAE National Fuels & Lubricants Meeting, Tulsa, Nov. 7, 1945.]

THE lubricating possibilities of silicones are summarized by the authors as follows:

1. Silicone fluids offer possible solutions to lubrication problems involving heat stability, oxidation resistance, nonvolatility, and low change of viscosity with change in temperature.

2. Different types of silicone fluids vary in their lubricating ability and in their behavior toward various rubbing metal surfaces. Some silicone fluids approach petroleum oils in ability to reduce wear.

3. Silicone greases, because of their oxidation resistance and low volatility, should find application in the operation of ball bearings under severe conditions and in permanently lubricated ball bearings where long service life is essential.

■ General Properties

Silicone fluids possess a combination of properties which go a long way toward eliminating some of the problems encountered by the lubrication engineer. Thinning out at high temperatures and the corollary thickening at low temperatures are effects which are greatly minimized in the silicone fluids. Fig. 1 compares the viscosity-temperature slopes of a family of silicone fluids with some typical petroleum oils, and the difference is quite apparent. For instance, if a silicone fluid and a petroleum oil of high viscosity index both have a viscosity of 50 centistokes at 100 F, then at 0 F the petroleum oil will have six times the viscosity of the silicone fluid.

Many lubrication problems involve a rather delicate balance between obtaining an oil with low enough viscosity and one with sufficiently low volatility. In other words, a kerosene with no tendency to evaporate would be very useful in some applications. Silicone fluids do not quite meet this specification, but it is possible to obtain nonvolatile silicone fluids in the viscosity range of 3 to 10 centistokes.

Low temperatures such as encountered in aircraft applications often force a compromise between pour point and volatility or between pour point and viscosity. That is, a petroleum oil of low enough volatility and high enough viscosity for a given application may have much too high a pour point. Table 1 shows that the DC 200 silicone fluids, which are nonvolatile and which can be obtained at viscosities of 1000 centistokes and higher, all have pour points below —50 F. Other silicone fluids, such as the DC 510, 500 centistokes, were developed for uses requiring a high-viscosity fluid with an even lower pour point (—96 F).

The utility of properties such as heat stability, nonvolatility, and oxidation resistance is illustrated in Figs. 2 and 3. Fig. 2 shows an apparatus designed to subject a conveyor chain to alternate high temperature (300 F) and high humidity, with a rest period between these two extreme conditions. The purpose of this test machine was to develop lubricants which would withstand this type of operation, and Fig. 3 shows one of the test results. After six weeks of operation, the length of chain lubricated with petroleum oil has rusted so badly that the rollers will no longer turn, largely because the high-temperature condition has oxidized and evaporated the petroleum oil. The chain treated with silicone fluid (DC 710) is still in excellent condition, shows almost no rusting and still

LUBRICANTS

by T. A. KAUPPI and W. W. PEDERSEN
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THE AUTHORS: T. A. KAUPPI, manager of product engineering of Dow Corning Corp., has been instrumental in engineering for industrial application a wide variety of silicone products including fluids, oils, greases, and so forth. His previous connections have been with the Institute of Paper Chemistry and Dow Chemical Co. Mr. Kauppi is a graduate of Brown University. W. W. PEDERSEN received his technical training at the University of Louisville, after which he joined the Cellulose Division of the Dow Chemical Co. as product development engineer. In 1942 he was assigned to the product development of silicones.

has a protective film of the silicone lubricant.

The properties mentioned above, low viscosity-temperature slope, nonvolatility, and low freezing point, are all very desirable in a lubricant. When these properties are combined with the heat stability, oxidation resistance, and general chemical inertness characteristic of silicones, the obvious conclusion seems to be that silicone fluids would make almost perfect lubricants for any sort of application. However, the one property of silicone fluids which has not yet been mentioned is that of lubrication; the above conclusion as to utility is valid only if the silicone fluid reduces friction and prevents wear between the moving surfaces involved.

■ Lubricating Properties

One of the first experiments carried out on the lubricating properties of silicone fluids involved pumping the fluid with a small gear pump at various outlet pressures. The pump was operated at approximately normal rated speed. An initial test on an SAE 10 motor oil showed that no appreciable wear occurred during 10 days of operation at outlet pressures ranging up to 100 psi. A run made with DC 200 silicone fluid of comparable viscosity produced extreme wear, as shown by suspended particles of steel in the fluid, within 1 hr of operation at 25 psi outlet pressure. When the pump was disassembled, it was found that the wear had occurred mostly in the steel bushings supporting the gears, and between the gears and the steel side-plates; apparently very little wear had occurred between the gears themselves. Several of these gear pumps, of various sizes, were being used regularly in the manufacture and processing of silicone fluids, and it was found that no trouble with wear had been encountered. However, the pumps were operated at considerably lower than normal rated speeds, and the outlet pressures rarely exceeded 100 psi.

It was apparent from the above and from similar experiences in other types of equipment that the DC 200 fluids did not have the lubricating properties of an ordinary petroleum oil. DC 200 fluids of various viscosities were being used at that time for instrument lubrication with considerable success, but whenever they were tested in applications involving high rubbing speeds or high unit bearing pressures, the results were unfavorable. The conclusion was reached that the DC 200 type of silicone fluid has no extreme-pressure lubricating properties, and actually begins to fail at unit pressures considerably lower than those normally accepted as the beginning of the extreme-pressure range.

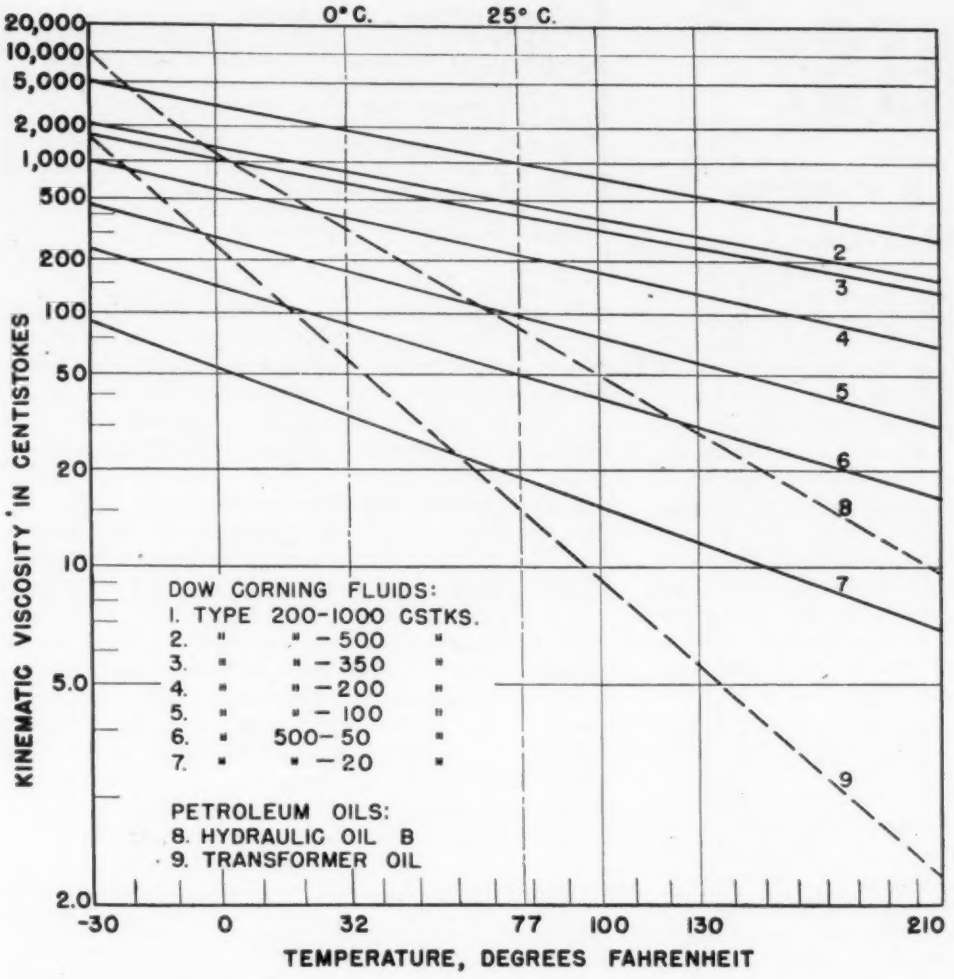
■ Wear Test

A few trials on the available lubricant testing machines showed that they were not designed to measure lubricating properties in the pressure range in which DC 200 began to fail. Therefore, a simple device was set up to permit measurement of wear between sliding surfaces over a range of rather low pressures. As shown diagrammatically in Fig. 4, this lubrication tester merely slides a ¼-in. metal

Table 1 – Physical Properties of Silicone Fluids

Uses of Liquid Silicones	Fluid Type	Viscosity Grade: Centistokes at 25C	Viscosity Grade: SSU at 100 F	Viscosity Temperature Coefficient $\left(1-\frac{V210\ F}{V\ 100\ F}\right)$	Freezing Point: C	Freezing Point: F	Boiling Point: Temperature, C	Boiling Point: Pressure, mm Hg	Flash Point, minimum F	Specific Gravity, 25C/25C	Lb per Gal at 25C	Expansion Coefficient k x 1000 per C 25C to 100C	Refractive Index at 25C
Damping Fluids		0.65	29	0.31	−68	−90	99.5	760	30	0.761	6.35	1.598	1.3748
Hydraulic Fluids		1.0	30	0.37	−86	−123	152	760	100	0.818	6.84	1.451	1.3822
Instrument Liquids		1.5	31	0.46	−76	−105	192	760	160	0.852	7.11	1.312	1.3872
Liquid Dielectrics	DC 500	2.0	32	0.48	−84	−119	230	760	195	0.871	7.27	1.247	1.3902
Foam Inhibitors		3.0	34	0.51	−70	−94	70–100	0.5	225	0.896	7.48	1.170	1.394
Mold Release Agents		5.0	40	0.55	−70	−94	120–160	0.5	270	0.918	7.67	1.095	1.397
Waterproof Treatment for		10	52	0.57	−67	−88	>200	0.5	350	0.940	7.85	1.035	1.399
Glass and Ceramic Insulators		20	78	0.59	−60	−76	>200	0.5	520	0.950	7.93	1.025	1.400
Special Lubricants for		50	198	0.59	−55	−67	>250	0.5	540	0.955	7.98	1.000	1.402
Glass, Rubber, and Plastics					ASTM Pour Point*		Volatility on Heating 48 Hr						
					C	F	at C	%					
		100	385	0.60	−55	−67	200	<2	600	0.968	8.08	0.969	1.4030
		200	785	0.62	−53	−63	200	<2	615	0.971	8.10	0.968	1.4031
	DC 200	350	1340	0.62	−50	−58	200	<2	625	0.972	8.11	0.966	1.4032
		500	1850	0.62	−50	−58	200	<2	625	0.972	8.11	0.965	1.4033
		1000	3690	0.62	−50	−58	200	<2	640	0.973	8.12	0.963	1.4035
	DC 510	50	177	0.62	<−70	<−94	250	<10	530	0.986	8.21	0.95	1.423
		500	1710	0.66	−72	−98	250	< 5	600	1.001	8.34	0.97	1.427
High-Temperature Baths	DC 550	50 to 75	150 to 213	0.77	−52	−62	250	<10	530	1.054	8.80	0.78	1.500
General Lubrication at High Temperature		200	485	0.86	−22	−8	250	<32	480	1.095	9.14	0.73	1.530
	DC 710	350	785	0.88	−17	+1	250	<22	500	1.103	9.20	0.72	1.535
		500	1060	0.89	−12	10	250	<15	520	1.112	9.27	0.71	1.539

* DC 200 fluids are not recommended for use below about −40 C (−40 F).



■ Fig. 1 – Viscosity-temperature slopes of liquid silicones compared to petroleum oils

ball over a flat metal strip. The ball is held rigidly in a chuck which can be vertically loaded in the range of 500 g to 4000 g, and which has a horizontal reciprocating motion. The amplitude of the oscillation is 2⅜ in., and the speed is 72 complete oscillations per minute. The flat metal strip is held stationary in a pan which contains the oil to be tested, and the usual method of evaluation is to weigh the test strip before and after a 2-hr run. The loss in weight of the test strip, or in some cases the depth of groove worn in it, measures lubrication efficiency.

A few preliminary runs were made on the wear tester to determine reproducibility of results and the effect of varying loads on the amount of wear. Fig. 5 shows the results obtained when a steel ball bearing was used in the chuck and a strip of shim brass was the wearing surface. The tester was operated for 2 hr and the weight loss of the brass strip plotted against the load applied to the steel ball. It is apparent that the wear obtained with DC 200 silicone fluid is greater than with an SAE 10 motor oil. Similarly, Fig. 6 shows the effect of running time of the tester on weight loss of the brass strip. The wear obtained with the petroleum oil in 16 hr is about the same as that obtained with DC 200 in ¾ hr.

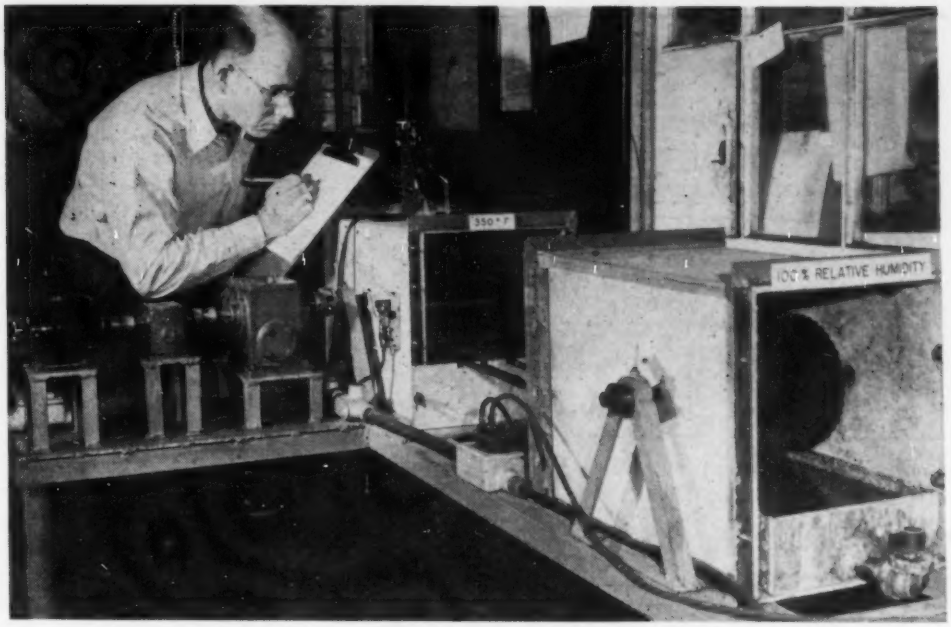
Since the results were reproducible and explained why difficulties had been encountered in the use of silicone fluids, the wear tester was used to evaluate the lubricating properties of a large number of experimental silicone fluids. It was found that variations in the chemical composition of the silicone fluids had a great effect on lubricating properties, and several new types were developed.

The wear test results on one of the newer silicone fluids are also shown in Figs. 5 and 6. DC 710, in Fig. 5, shows less wear up to 2000-g load than SAE 10 petroleum oil, but about twice as much wear at 3000-g load. Fig. 6 shows that at 1500-g load, DC 710 produces considerably less wear than the petroleum oil at all running times up to 16 hr. The tentative conclusion to be drawn from these results is that DC 710 is better than petroleum oil for steel-to-brass lubrication at moderate loads, but is less effective than petroleum oil at loads near the e-p range.

■ Comparative Wear Tests

The wear test results shown in Figs. 5 and 6 were obtained with a steel ball and a brass test strip. For the purpose of evaluating relative lubricating ability of various silicone fluids, however, results were obtained with a steel ball bearing and a soft steel test strip. This method of testing is much more severe, since a DC 200 silicone fluid shows 5 to 10 times as much wear when a soft steel strip is used in place of the brass strip.

Table 2 shows wear test results, using a hardened steel ball and a soft steel strip. The tests were run for 2 hr at



■ Fig. 2 – Apparatus designed to test lubrication life of silicone fluid DC 710 and petroleum oils on conveyor chains cycled through temperature chamber held at 300 F and humidity chamber with rest periods at room temperatures intervening

1500-g load and at 4000-g load. Distilled water, petroleum solvents, and petroleum lubricating oils all show relatively little wear, and of course the petroleum motor oil is the best of the group. Among the silicone fluids, DC 200 and DC 510 are quite poor, DC 550 is somewhat better, and DC 710 seems to be equivalent to petroleum oil of about the same viscosity.

Table 2 – Comparative Wear Tests on Fluids
(2-hr test)

Fluid	Wear, 0.0001 g	
	1500-G Load	4000-G Load
Distilled Water	13	10
Petroleum Naphtha	20	56
Kerosene	16	40
Hydraulic Oil	8	9
SAE 20	0	3
DC 200	740	..
DC 510	700	..
DC 550	330	..
DC 710	0	2

■ Wear on Various Metals

Table 3 shows the results obtained when a hardened steel ball slides on various metal surfaces. It is apparent that aluminum has the greatest tendency to wear under these conditions, and spring steel the least. DC 710 is a definite improvement over DC 200, and appears equivalent to SAE 20 except in the case of aluminum. It should be noted that test values below 5 are all equivalent, since this is the order of accuracy of the test procedure.

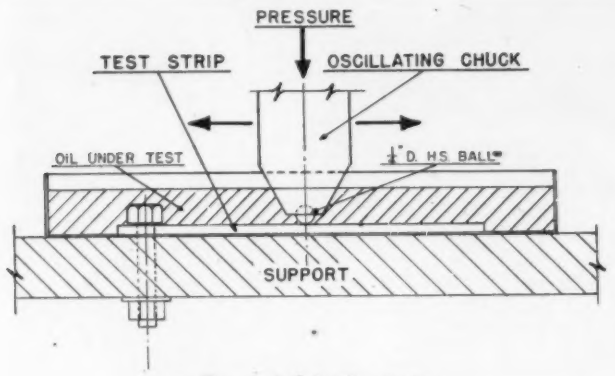
Table 3 – Wear Test on Metal Combinations
(2-hr test; 1500-g load)

Hardened Steel Ball against:	Wear, 0.0001 g			
	SAE 20	DC 200	DC 550	DC 710
Brass	4	130	40	3
Aluminum	65	250	430	300
Magnesium	10	40	45	4
Soft Steel	0	750	330	0
Hardened Spring Steel	2	4	3	3

It is interesting to note the relative behavior of DC 200 with soft steel and spring steel, and to compare this with results obtained on the pumping test. The gear pump cited previously had soft steel sleeve bearings supporting the gears, and soft steel side plates, and excessive wear occurred on these parts. Almost no wear was found on the gears themselves. Another test, using a rotary vane-type pump, has been operating with DC 200 with excellent results. In this case, the steel parts are hardened, and the side plates are of bronze. The pump has been running for over 2000 hr at 1000 psi pressure, and with the silicone fluid held at a temperature of 90 C. The test is being continued, and to date the wear is negligible.

The data discussed above show that considerable differences in lubricating ability exist between the various silicone fluids, and that at least some of them are suitable for a range of lubrication applications. Even the DC 710 fluid, however, is not an extreme-pressure lubricant. Since silicone fluids are totally different from petroleum oils, it is almost impossible to predict from petroleum oil experience how a particular metal combination will behave with silicones; in general, a preliminary test will quickly show

■ Fig. 3 – After six weeks of operation on test apparatus shown in Fig. 2, upper chain, lubricated with petroleum oil, is so badly rusted that the rulers are frozen, while lower chain, lubricated with DC 710, is still flexible, free of rust, and operable



■ Fig. 4 – Lubrication tester

whether one of the silicone fluids is suitable from the standpoint of preventing wear.

■ Silicone Greases

Many grease problems are caused by the tendency of the oils used in greases to evaporate or oxidize, resulting in eventual hardening or caking of the grease. Since silicone fluids are definitely superior in volatility characteristics and oxidation resistance, work was started about two years ago to develop a silicone grease for ball bearings.

The first silicone bearing grease developed, DC 41, was compounded of a finely divided carbon black as a thickener and a silicone fluid. Because it is an almost completely inorganic grease, DC 41 does not melt at temperatures in excess of 480 F, but more frequent lubrication is required because of increased bleeding at temperatures above 300 F. Its extreme heat resistance recommends it for special applications, such as the lubrication of pumps handling molten salts and oven machinery of all kinds. DC 41 is serviceable at temperatures as low as −4 F. In the temperature range of 200 to 300 F, DC 41 has between 5 and 10 times the lubricating life of high-quality petroleum greases designed for use at elevated temperatures. Tested in No. 212 bearings operating under a load of 150 lb at a speed of 1750 rpm, this grease was still serviceable after 2600 hr of operation at 257 F. In the same testing equipment at 347 F, DC 41 had a service life of 900 hr.

It was found that, although DC 41 performs well in ball bearings at 1750 rpm, it is not suitable for speeds in the range of 10,000 rpm. Another type of silicone grease was developed, thickened with a metallic soap, and found to give much better results at high speeds. This grease, DC 44, is now under test in a No. 204 bearing operating at 10,000 rpm and 300 F. To date the bearing has operated 1500 hr and shows no signs of deterioration, as measured by temperature rise and running torque of the bearing. DC 33 is a grease similar to DC 44, but designed for low-temperature use. It does not thicken at temperatures as low as −90 F and has given good test results in bearings operating up to 300 F.

Table 4 compares the volatility of the above three silicone greases with three petroleum greases recommended for high-temperature applications. The volatility of the silicone greases at 175 C is less than 5% in 40 hr, while the petroleum greases lost from one-third to over one-half of their original weight.

Table 4 – Percentage of Weight Lost after 40 Hr at 175 C (347 F)

High-Temperature Petroleum Greases, %			Silicone Greases, %		
A	*B*	*C*	DC 33	DC 41	DC 44
58.3	39.3	34.8	4.8	4.2	3.3

After 40 hr of exposure at 175 C a crust had formed over the surface of the three petroleum greases. The silicone greases were still soft and serviceable.

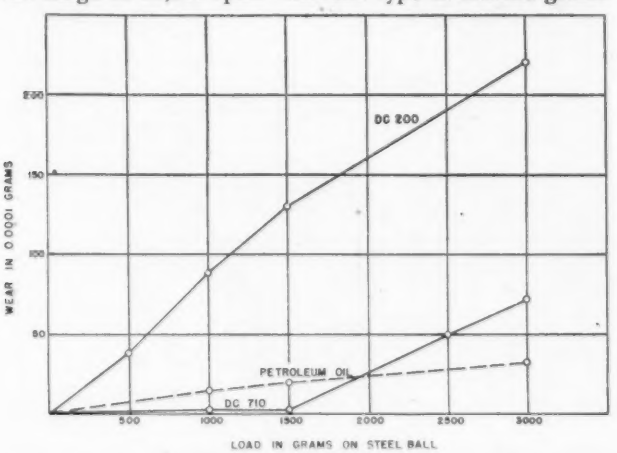
The volatility and oxidation resistance characteristics of silicone greases shown in Table 4, combined with the operating results obtained to date in ball bearings, indicate that silicone greases have a logical application in the permanent lubrication of ball bearings. Sealed ball bearings with no provision for relubrication are receiving increased attention because of their obvious operating advantages and the use of silicone greases should greatly increase the service life of such units.

■ Conclusions

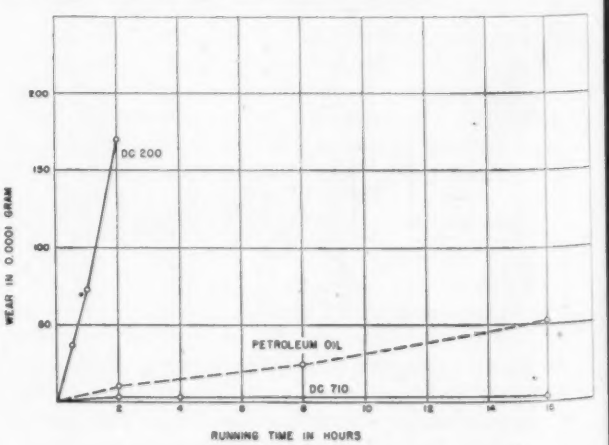
1. Silicone fluids may solve lubrication problems involving heat stability, oxidation resistance, nonvolatility, and low change of viscosity with change in temperature.

2. Different types of silicone fluids vary in their lubricating ability and in their behavior toward various rubbing metal surfaces. Some of the silicone fluids approach petroleum oils in ability to reduce wear.

3. Silicone greases, because of their oxidation resistance and low volatility, should be applicable to ball bearings under severe conditions and in permanently lubricated ball bearings where long service life is essential.



■ Fig. 5 – Wear tester results for hardened steel against brass – wear versus load, 2 hr elapsed time



■ Fig. 6 – Wear tester results for hardened steel against brass – wear versus time, 1500-g load

Aviation Fuels—What the Airlines WANT and EXPECT

by GILBERT K. BROWER
Chief Materials Engineer
American Airlines, Inc.

AS the representative of an industry which will have, within 10 years, an interest in approximately 1,000,000,000 gal of aviation fuel annually, I will attempt to point out what the airlines want and expect in the line of aviation fuels at as early a date as possible. Reference will be made to current specification requirements and, along with this, the quality desired by the airlines will be presented, with reasons therefor; allusion will then be made to current work being conducted on a cooperative basis by the Coordinating Research Council's CFR (Coordinating Fuels Research) Committee as well as some recent fuel flight tests conducted by the commercial airlines, and finally, a brief resumé of the grades which will probably be required in the future with due consideration to the economics involved.

First, in order to get some idea of the potential volume involved and the relative costs to the air transport operators, the following data should be enlightening. Fig. 1 illustrates the past, present, and estimated future consumption of aircraft fuels by the domestic airlines from 1926 and projected through 1954, as based on available CAA figures. Add to the 510,000,000 gal estimated demand by the airlines in 1954 an estimated 635,400,000 gal by other civil aircraft, and the total is over 1,000,000,000 gal annually. In a paper presented before the ASTM in 1940, the author estimated that the airlines' consumption of fuel by 1951 would approximate a half billion gal a year. This was before the war, and yet that figure is quite consistent with the present CAA prediction, so we do not feel that they are overly optimistic. This means then that civil aircraft will be consuming a tank car of fuel every 5 min, or at a rate of 2000 gpm.

What does this mean to the air transport operator? Table 1, data taken from a paper presented by Hoben,[1] indicates that fuel and oil, including taxes, account for over 10% of the total operating costs of an airline in passenger service, and in cargo service approximately 15%. On the same basis, fuel and oil account for 30 to 35% of the total direct flying cost, and flying operation for 48 to 67%. These figures were based on an actual average cost of 14.5¢ per gal for fuel used during 1943 when bearing the burden of fuel consumed in nonrevenue flying, ground testing, and so on, but charged only against revenue flights. For the DC-4, a 3¢ per gal premium was added arbitrarily as a contemplated difference between the 91-octane grade fuel used in DC-3's and the probable 100-octane fuel required for the DC-4's. Thus, the annual cost to the air transport operators for fuel and oil alone will exceed $70,000,000, and each penny a gallon difference can mean an approximate $5,000,000 increase or decrease. You can figure what this means when compared to the profits normally made by the airlines.

■ Specification Requirements

Now to get down to the substance of my topic, that is, what do the airlines want in the way of quality for their aviation fuel requirements?

The question of fuel quality can be generalized in several different ways, but I rather prefer to make a specific approach to the problem and hence must, of necessity, have some past datum point for comparison. With this in mind, use is made of the current AN specifications for Grade 91/96 aircraft-engine fuel (restrictions hold true for higher grades) and of ASTM Tentative Specifications for Aviation Gasolines – Designation D-615-41T issued in

WITHIN 10 years, Mr. Brower predicts, airlines will use over 1,000,000,000 gal of fuel annually, consequently, what the airlines want and expect in the way of aviation fuels is of major importance.

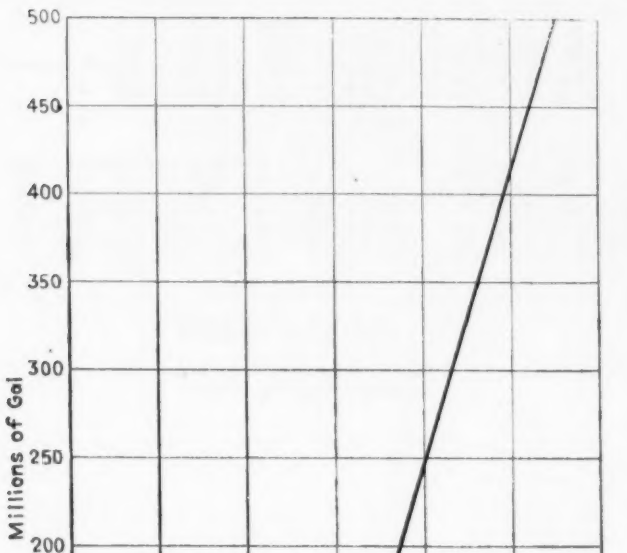
The author stresses particularly that airlines would like less talk of so-called "super fuels," especially at premium prices, and more interest in down-to-earth, realistic development of economical and safe fuels of good quality.

After presenting a resumé of the grades of fuel that he expects will be required in the future, Mr. Brower expresses the thought that to obtain these results, the cooperation that has developed between the fuel refiner, the engine and airframe manufacturers, and the air transport operators must be maintained.

THE AUTHOR: GILBERT K. BROWER (M '35), an American Airlines associate since 1931, is now chief materials engineer in charge of chemical, physical and metallurgical laboratory tests and control of materials and processes. Mr. Brower, who has degrees from Antioch College and the University of Pittsburgh, worked for several of American Airlines' predecessors prior to being headquartered in New York.

[This paper was presented at a meeting of the Metropolitan Section of the SAE, New York City, April 5, 1945.]

[1] See "Cargo Plane Design from Operator's Standpoint," by H. E. Hoben. Presented at SAE National Air Cargo Meeting, Chicago, Dec. 4, 1944.



■ Fig. 1 – Fuel consumed in scheduled airline operations (actual through 1944, estimated to 1954) – based on CAA figures

1941, last published, we believe, in the 1942 book of ASTM Standards, and discarded, *please note,* without the recommendations of a single air transport operator, in 1944. Tables 2 and 3 give specification requirements for reference, and from this point on, each item will be reviewed in detail.

Materials and Workmanship – It is aphoristic that fuels should be free from water, sediment, and suspended matter, and should consist of hydrocarbons with known characteristics. In this connection, therefore, considerable thought must be given to some method of wording or specification requirement which will prevent the use of any hydrocarbon with known unsatisfactory properties, such as questionable olefins. At present this is not as simple as it seems but some answer must be found.

Freezing Point – A maximum of −76 F is indicated. For conventional, present-day aircraft fuels, this requirement has not been restrictive and has proved satisfactory in service; this is to be expected, since no drop below this temperature would be expected in a conventional carburetor. No change is desired.

Water Tolerance – The current requirements that the fuel be substantially immiscible with water and separate promptly without a volume change exceeding ±2 ml has been entirely agreeable to all concerned and needs no further comment or desired changes. This requirement is particularly applicable to those fuels to be stored over water.

Reid Vapor Pressure – At present all fuel is purchased to meet a 7-psi maximum Reid vapor pressure and, from the airlines' viewpoint, no relaxation of this limit is desirable. It is naturally restrictive as regards the type or amount of certain otherwise satisfactory hydrocarbon components. Unless, however, this limit was so unduly confining that an overall increased fuel cost would justify an extensive research program and alteration of the fuel system of present transport aircraft, no change can be condoned.

Sulfur – Current and past specifications set a maximum limit of 0.05% as the sulfur content of aviation fuels. At no time in many years have the laboratories of two of the major airlines seen a sample of aviation gasoline which contained over 0.05%; therefore, there is no adequate background of experience to state what point above such a limit would be permissible. Engine manufacturers who have had experience with fuels of higher sulfur contents state that they leave a substantial volume of deposits with a definite sulfur color, which occurs at moderately low operating carburetor air temperatures. With newer methods of refining, there has been some hint that in certain instances present restrictions can be unduly harsh and the limit may require some relaxation. This can only be answered by the fuel refiner and the economics involved in distribution. If there is any considerable volume involved, then a waiver or relaxation of quality must be considered. At the moment, no change is desirable.

Color – Recent color requirements for leaded aviation fuels have invariably called for the addition of a blue dye to match a certain color shade or intensity. Such a general designation when applied to aviation fuels of different grades is definitely unsatisfactory from the operator's viewpoint. Back in the days of 1930 we had a green color for 73-, blue for 80-, and purple for 87-grade octane fuels. It is a "must" from an airline safety factor, that something be done in the future to make color coding of fuels for various knock-rating grades mandatory. Here is one point where there would be no question of governmental regulation since the petroleum industry and the manufacturer of tetraethyl lead have been most uncooperative despite repeated requests. The airlines want identifying colors used for the various fuel grades. In so doing, we do not want any more dye than now used in the present 91 fuel.

Table 1 – Distribution of Operating Expenses (Percentages Based on ¢ per Revenue Mile)

	DC-3 Passenger (Actual-1943)	DC-3 Cargo	DC-4 Cargo
Flying Operations (except Fuel & Oil)	11.3	16.2	7.3
Fuel and Oil – including Taxes	10.5	14.3	14.9
Flight Equipment Maintenance – Direct	8.7	11.9	10.8
Depreciation – Flight Equipment	3.5	5.1	8.5
Total Direct Aircraft Operating Expense	34.0 (Fuel 31%)	47.5 (Fuel 30%)	41.3 (Fuel 35%)
Ground Operations	20.2	15.7	17.2
Ground Equipment Maintenance – Direct	0.9	1.2	1.1
Equipment Maintenance – Indirect	6.4	8.6	7.8
Depreciation – Ground Equipment and Property	1.8	2.5	3.5
Passenger Service	8.2	....	....
Cargo Handling, Billing, and Insurance	....	4.4	6.1
Traffic and Sales	12.1	6.1	9.0
Advertising and Publicity	4.9	4.0	4.0
General and Administrative	11.5	10.0	10.0
Total Ground and Indirect Expenses	66.0	52.5	58.7
Total	100.0	100.0	100.0
Based on ¢ per Revenue Ile	86.91	59.83	122.90

Table 2 – Comparative Fuel Quality Requirements

	ASTM – D615-41T		AN-F-26	Quality Wanted		Test Methods		
Grade	91	100	91/96	91/96	100/130	ASTM	FSB*	Other
Freezing Point, F (max)			−76			—	141.1	—
Water Tolerance, volume change (max)			±2ml			—	325.1	—
Reid Vapor Pressure, psi (max)			7			D323-43	120.11	—
Sulfur, % (max)			0.05			D90-41T	520.13	—
Color		Blue		Blue	Special	—	—	As specified
Permissible Gum Inhibitors	1 lb max per 5000 gal as specified			Preferably none		—	—	—
Corrosion:								
Copper Strip at 212 F	No gray or black		—	No gray or black		D130-30[b]	530.31	—
Copper Dish, 100 ml	No gray or black and 5 mg max residue					—	330.11[c]	—
Acidity of Distillation Residue	Negative		—	Negative		—	510.2	—
Net Calorific Value, Btu per lb (min)	18,700			18,900		D240-39	250.2	—

[a] FSB methods as included in current revision of Federal Specification No. VV-L-791.
[b] Except temperature of 212 F.
[c] Alternate FSB Method 530.11 acceptable.

and it must be stable. Clear base fuels should have a color of +21, minimum, since we have no definite satisfactory experience on fuels not meeting this requirement.

Gum Inhibitors – This is a moot question which cannot successfully be answered by an air transport operator. Certainly as an industry, we would most heartily welcome any improvement in our fuel quality which would reduce the necessity of our contaminating good fuel with inert ingredients of no final value as regards the operation of our aircraft engines. These additives have many times in the past been the cause of suspicion, and justly so, as regards deposits in the induction systems of aircraft, in storage tanks, and of most importance in the engine proper. At the present time we must depend upon the good faith of our fuel vendor and the manufacturer of any given inhibitor as to what service experience may be expected. Based on past performance, only those listed are approved. We fully realize there may be several others equally as effective, if not more so, that are not as yet approved but must await results of long-time stability tests or satisfactory control tests. With this in mind, the operator is faced with the alternative of relaxing present stability requirements in favor of fuel with no inhibitor added. I might add that this is being given careful consideration and may result in two differing specifications dependent upon the ultimate storage location and rate of use of such fuels. There is not a great deal of question as regards the quantity limitation of permissible gum inhibitors; such additions must be kept to as low an amount as possible.

Corrosion – Normally, in the past, two standard procedures have been used as specification requirements to control the free sulfur and corrosive sulfur compounds in fuels, namely, the copper dish and the copper strip methods. To our knowledge, neither of these items has been thought restrictive, although the question always arises as to the need for including both methods, since they are of a qualitative nature only. The primary reason for including both requirements is based on the fact that in the copper dish test it has sometimes been possible to get only a peacock coloration which would pass the sample, but at the same time a copper strip would show an unsatisfactory gray or black discoloration. The copper dish test, although not satisfactory as such, may also be used as an indication of gum content. It may well be that such deposits forming during the test prevent the higher boiling constituents, which may be the cause of discoloration, from doing so in the copper dish. Both methods, therefore, are desirable with a requirement that no gray or black discoloration be permitted.

Acidity of Distillation Residue – This item has always been and should in the future be specified in order to act as a safeguard against fuels leaving the refinery and being accepted for service which have not been satisfactorily treated or neutralized against acid contamination in particular. It will always be expected that the distillation residue shall not be acid.

Calorific Value (Heat of Combustion) – In the past, specifications have almost invariably called for a minimum net calorific value of 18,700 Btu per lb. There has not been a very great deal of question as regards receiving fuel meeting this limit. Naturally, it is restrictive but not unduly so. Frankly, our desire would be to get a fuel having a minimum of 18,900, since fuel consumption varies almost directly as the Btu's. It is to be presumed that, dependent upon the postwar supply picture and economic considerations, some adjustment or compromise will have to be made relative to this figure. Certainly it will not be less than the present figure of 18,700; we hope it will have at least an 18,800 minimum and would prefer 18,900.

Tetraethyl Lead Content – Normally, specifications have limited the amount of tel to a maximum of 4 ml per gal. During the war, however, this has been raised to a maximum of 4.6 ml per gal in order to extend or increase the available supply of aviation fuels. The operators, of course, would prefer fuel without any tel whatsoever, since it builds up excessive deposits in the crankpins, propeller domes, supercharger clutch mechanism, cam gears, combustion chamber, and valves, as well as being a prolific source of potential corrosion headaches and storage deposits. The question is continually raised as to the premium value which could be paid per gallon for, say, reducing the tel content from 4 ml to zero by increments of 1 ml. Assuming that each milliliter of tel per gallon costs approximately a quarter of a cent or a total of one cent per gallon for fuel containing 4 ml, we feel that the economy effected by a reduction in tel would be more than offset by the difference or increase required to produce fuels from the improved base stocks which would be required.

Frankly, the value of tel reduction could not be determined without going to very extensive tests on a whole fleet of airplanes for at least an engine overhaul period and using fuels containing definite and known amounts of tel in regularly reduced increments. In this connection, it might be pointed out that in a normal engine overhaul period of 800 hr, approximately *40,000* gal of fuel is consumed. If a clear fuel costing a penny a gallon more than our currently leaded fuel was used, it would have cost us *$400* more. The average overhaul cost of an engine is approximately $900. If, therefore, the overhaul time was

Table 3 – Comparative Fuel Quality Requirements

	ASTM – D615-41T		AN-F-26	Quality Wanted		Test Methods		
Grade	91	100	91/96	91/96	100/130	ASTM	FSB[a]	Other
Tel, ml per U. S. gal (max)	4	3	4.6	4.0		D526-42	550.1	—
Gum (Residue), Actual, mg (max)	—	—		5		—	330.11[b]	As specified
Gum (Accelerated), mg per 100 ml (max)		6.0 (5.0 precipitate)		6.0 (5.0 precipitate)[c]		—	335.1	As specified
Acid Heat, F (max)	—	—	—	10		D481-39	350.1	—
Knock Value[d], Lean (min)	91	100	91	91.0	100.0	—	—	CFR-F3
Knock Value[d], Rich (min)	—	—	96	98.0	130.0	—	—	CFR-F4
ASTM Distillation, F (max)						D86-40	100.15	—
10% evaporated		158	167	158[e]				
50% evaporated		212	221	221				
90% evaporated		257	284	257				
End point		—	356	338				
Sum of 10 and 50% (min)		307	307	307				
Recovery, % (min)			97					
Residue, % (max)			1.5					
Loss, % (max)			1.5					

[a] FSB methods are included in current Federal Specification VV-L-791.
[b] Alternate FSB method 530.11 acceptable.
[c] Under special test procedure and no catalyst 16 hr instead of 5.
[d] New procedure covering entire *F/A* mixture range preferred when available.
[e] There may be a desire for a minimum on this point also – say 140 F.

doubled which is improbable due to elimination of tel alone, say to *1600* hours, then at a penny a gallon increase for a nonleaded fuel, the additional fuel cost would be $800 as against elimination of *one* overhaul at $900 each. *Thus, there would only be a gain of $100. This slight gain could probably not offset other added costs occasioned by doubling overhaul periods.* Regrettable to say, therefore, it looks for some time to come that, from an economic standpoint alone, the airlines must still be satisfied with a 4 ml maximum tel content. Here's a chance for some petroleum refiner to do a real job of salesmanship and service – it will pay dividends – provide lead-free fuel with no price increase.

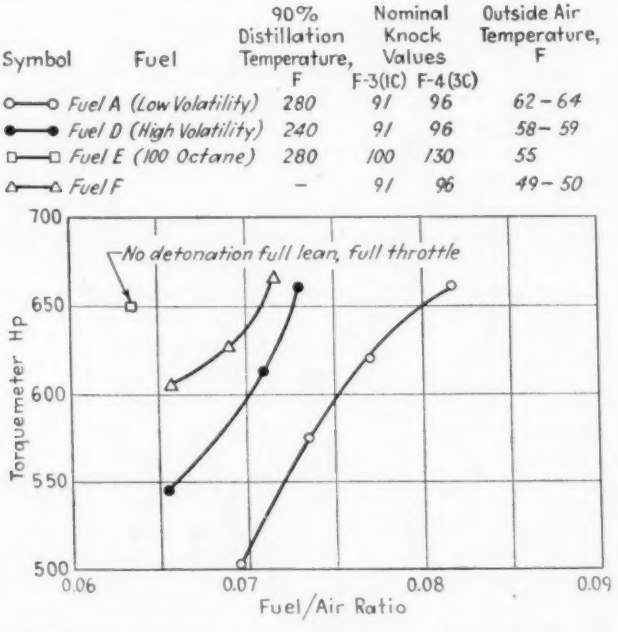
Gum (Actual) – Sometimes called copper dish gum or residue, this requirement has always in the past called for a maximum residue of 5 mg per 100 ml of fuel. It serves as a good indication of the actual amount of gum deposition which may take place in service when the gasoline tested is to be used immediately. True, in this test there are many imperfections; for instance, any other nonvolatile material will also show up, such as lubricating oil and dyes. As a rough quality requirement, however, and at least until a better method is evolved, it is quite imperative and desirable that this requirement be retained.

Gum (Potential – Oxidation Stability) – Oftentimes referred to as accelerated gum, previous specifications have always called for a maximum permissible content of 6 mg per 100 ml of sample after correcting for tel. This requirement is included to enable the consumer to estimate the tendency of the gasoline to form gum or undesirable lead precipitates in storage. Based primarily on the experience of, and research work done by, J. T. Hendren, chief chemist of Pan American Airways, Inc., it is quite apparent that the future thinking relative to this problem will necessitate a considerable amount of study and further investigation before a final satisfactory decision can be arrived at. In fact, the end result may be the issuance of a basic specification with two alternate accelerated gum requirements to permit either an inferior fuel, as regards oxidation stability, to be purchased for use with certain restrictions or a well stabilized fuel for those operators or areas where stability is of paramount importance and definitely required.

Certainly there is no desire on the part of the operators to relax the present specification limits but rather the trend will be toward more severe requirements, particularly so as regards those air transport concerns and areas requiring the best available fuel. We feel the following paragraphs written by Mr. Hendren amply, yet concisely, tell the story:

"In all leaded gasolines, there are always small amounts of tel decomposition products, mostly soluble, but including minute amounts of precipitates. This may easily be confirmed by the inspection of any aviation gasoline storage tank. In these concentrations, the gasoline is entirely satisfactory. When oxidation inhibitors, natural or added, are expended, the rate of decomposition increases sharply. Both lead precipitation and gum formation are autocatalytic and the gasoline becomes unfit for use in a fairly short time.

"A multitude of factors are involved, such as storage temperature, relative freshness of ethyl fluid when added at the refinery, amount of 'natural inhibitors' present, absence or presence of water in storage containers, contaminants present, accessibility of air in storage, exposure to light, amount of 'handling' received prior to 'permanent



■ Fig. 2 – Part-throttle cruise; knock-limited conditions; test procedure 2*A* – 1850 rpm, 4000-ft altitude, full cold carburetor air temperature

storage, and the like. All exert considerable effect upon stability and preclude the possibility of a simple routine test to predict it.

"It is, therefore, our conclusion that an oxidation resistance test, that is, a variation of the present bomb test, is the best available approximation to use for stability characteristics. For simplicity and uniformity, we feel the present test with extended oxidation time would be satisfactory. Sixteen hours at 100 C and 100 psi oxygen does not appear unreasonable, as some of the present fuels have so been run with no lead precipitation and less than 5 mg of glass dish, steam oven gum (no catalyst)."

Acid Heat – Here is one item which can always be expected to be controversial. A requirement limiting the maximum acid heat temperature as a means of restricting the use of certain unsaturated hydrocarbons has been in and out of aviation gasoline specifications many times. While it is true that there is no really valid knowledge as regards this want, it is known to exclude fuels which are recognized as being undesirable, but, does it also limit or exclude fuels having advantageous characteristics? For instance, in former days when knock ratings were run using an Ethyl Corp, Series 30 engine, this was the only means by which it was readily possible to exclude certain unsatisfactory, thermally cracked fuels. These fuels would otherwise have been accepted because some unsaturated hydrocarbons contained therein were appreciated in knock value by this method of test out of proportion to other fuels, yet would prove unsuitable with respect to detonation in full-scale engines or in storage stability.

New methods of determining knock ratings eliminate this problem but now the question is, whether or not, with all the studies pertaining to oxidation stability, poor fuels cannot still be accepted unless this requirement is reinstated. As to the exact limit to set, there is some uncertainty. Prior to 1940, however, 10 F did not appear restrictive, if one accepted the premise that competitive prices are indicative of supply. Until, therefore, the postwar picture manifests a supply limitation, 10 F is the limit the airlines must logically request. Only the refiners can tell us what we may expect, based on the economics presented by the possible retention of certain of the newer refining units and processes.

Knock Value – Only the engine is able to tell us what knock value it requires for satisfactory operation under any given set of conditions. It, therefore, is the prime responsibility of the engine manufacturers to tell us, of the airlines, what any given engine model requires. Since this is really a grading requirement and knock-limited performance of aviation fuels will further be discussed later on in this article when flight tests and other fuel characteristics are discussed, only knock test methods and data desired will be referred to at this time.

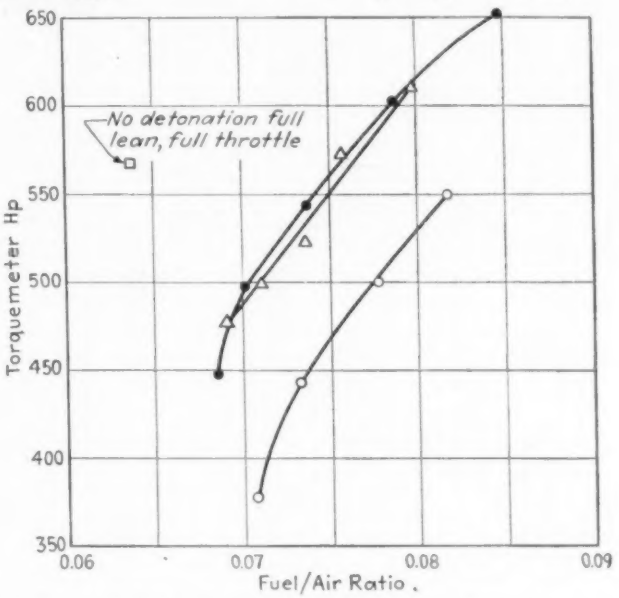
Perhaps the requisite factor most needed at this time is some improved method of determining knock values which will cover not only the entire expected operating range of the fuels being used as regards fuel/air ratios but also one which is more consistently and closely related to flight detonation properties. The use of the present CFR F-3 (1C) and F-4 (3C) methods has helped considerably in this respect, as we then have both a lean and rich mixture rating; however, not being satisfied with stopping short of the goal the airlines want, and expect, in the near future, to have a method and rating which will cover the whole range of fuel/air values.

One other requirement which must be clarified as soon as possible is the minimum tolerance in ratings currently being accepted. Speaking for American Airlines, Inc., I can state that in any future specification, we shall, for instance on a 91/96 grade, expect 91.0 as the absolute minimum, not 90.6.

Distillation Range – Now I come to a subject which has been and always will be subject to all types of arguments. Prior to the war, airline specifications called for the following maximum distillation temperatures: 10% evaporated: 158 F, 50% evaporated: 212 F, and 90% evaporated: 257 F, whereas the military services called for 167 F, 212 F, and 275 F, respectively, with the current requirements permitting a 50% evaporated limit of 221 F and a 90% of 284 F. As is well known, the 10% point is normally included as a quality criterion in order to control the starting characteristics of the fuel in that it can be correlated to atmospheric temperatures at which a given fuel will vaporize sufficiently under manifold (air equilibrium) conditions to start a cold engine.

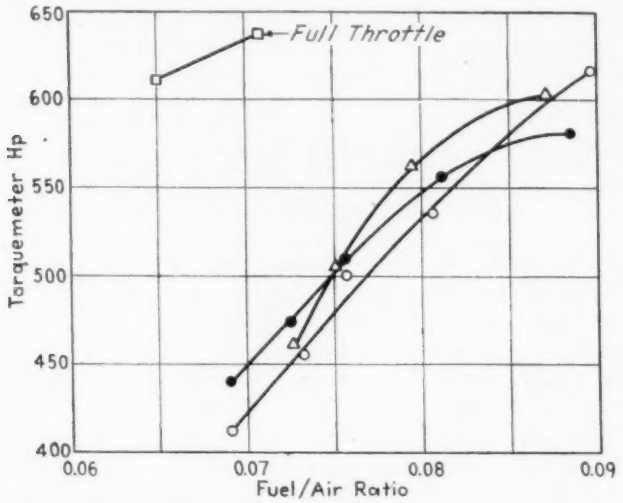
In this connection, it must be remembered that, since only vaporized fuel will burn and since only the more volatile portions vaporize at low temperatures, the pilot must increase the fuel/air ratio to produce a combustible mixture for starting by priming and mixture enrichment. The 50% point indicates the warming up and throttle response characteristics of a given fuel, and hence controls the acceleration of an engine when warm. This requirement is essential in order to assure maximum take-off power or for other emergency power conditions, for when the throttle is suddenly opened, the equilibrium of vapor-

Symbol	Fuel	90% Distillation Temperature, F	Nominal Knock Values F-3(1C)	F-4(3C)	Outside Air Temperature, F
○—○	*Fuel A (Low Volatility)*	*280*	*91*	*96*	*60*
●—●	*Fuel C (High Volatility)*	*229*	*91*	*96*	*56-57*
□—□	*Fuel E (100 Octane)*	*280*	*100*	*130*	*53*
△—△	*Fuel F*	–	*91*	*96*	*50-51*



■ Fig. 3 – Part-throttle cruise; knock-limited conditions; test procedure 2B – 1850 rpm, 4000-ft altitude, 100 F carburetor air temperature

Symbol	Fuel	90% Distillation Temperature, F	Nominal Knock Values F-3(1C)	F-4(3C)	Outside Air Temperature, F
o—o	Fuel B (Low Volatility)	279	91	96	48-49
●—●	Fuel D (High Volatility)	240	91	96	48-49
□—□	Fuel E (100 Octane)	280	100	130	53
△—△	Fuel F	–	91	96	48-49



■ Fig. 4 – Part-throttle cruise; knock-limited conditions; test procedure 2E – 1850 rpm, 4000-ft altitude, 150 F carburetor air temperature

ized and liquid fuel in the inlet system is affected and may upset the F/A mixture sufficiently to interrupt engine operations momentarily. The 90% point prevents too great a percentage of less volatile fuels being present which may cause poor overall distribution of the fuel to the various cylinders with consequent adverse effects as regards power loss, temperature limitations, fuel consumption, and even detonation, depending upon operating conditions. The less volatile components may also have a tendency to remain liquid and hence wash the oil from pistons, cause undue dilution in cold weather, and so on.

The expectations are, therefore, that prewar commercial aircraft-engine fuel volatility characteristics will still be desired with, perhaps, a slight increase being allowed for the 50% limit. You can, therefore, expect a 10% limit of 158 F, 50% of 221 F, and 90% of 257 F, with other factors such as sum of 10 and 50% points, per cent loss, and residue remaining the same. A decrease in permissible loss would be of some advantage in cutting down fuel losses due to storage evaporation, handling, and so on. Further specific and illuminating references which will crystallize your thinking along these lines will be presented immediately in a review of some flight tests conducted by one of the major airlines.

■ Flight Investigations

In November of 1944, American Airlines, Inc., in order to attempt an evaluation of aviation fuels then being supplied, ran a series of flight tests at Phoenix and Tucson. Since the data obtained from these tests are most pertinent to this subject, I shall present them in considerable detail. Basically, there were three fundamental differences in certain of the fuel supplies then being furnished with which the air transport operators did not have to contend prior to 1941, namely, an increase in tel content from 4.0 to 4.6 ml per gal, an increase in the 90% volatility limit from 257 F to 284 F, and a tendency to receive most fuels with knock ratings ranging between 90.6 and 91.0 rather than between 91.0 and 92.0 as formerly. With this in mind, and since we had, during the summer of 1944, engine difficulties not definitely attributable to some known cause, flight tests were planned on fuels of basically three varying types: one a fuel of 91 octane with 1940 commercial airline volatility characteristics; another 91-octane fuel meeting essentially prewar Army and currently specified Army-Navy 91-octane fuel distillation limits, and a 100/130 grade fuel also with the less volatile limits. Table 4 lists the laboratory inspection data for these fuels.

All flight tests were conducted under the direction of M. G. Beard, director of flight engineering, and the ship was equipped with a full complement of test apparatus including torquemeters, flowmeters, and Sperry detonation equipment. Typical test results illustrative of what fuel characteristics mean are shown in Figs. 2, 3, and 4.

Fig. 2 shows a test which is typical of the major part of an airline's flight operations, that is, cruising at 4000-ft

Table 4 – Fuel Analyses – 1944 Flight Tests

Test Designation	C	D	DX[a]	B	A	AX[a]	E
Location	New York	Phoenix	Phoenix	Tucson	Tucson	Tucson	Tucson
AA Laboratory No.	414	424	443	425	422	442	426
Distillation, F							
Initial	120	120	114	113[b]	108	111	105[b]
10%	149	148	152	170[b]	154	156	143[b]
50%	177	158	194	214[b]	216	219	214[b]
90%	229	240	237	279[b]	280	278	280[b]
End point	287	274	272	317[b]	324	323	326[b]
Knock Values:							
F-3 (1C) Sewaren	90.7	90.7	—	90.2	90.0	—	97.9
F-3 (1C) Waukesha	91.9	90.6	—	90.9	90.2	—	99.4
F-3 (1C) Stancal	—	—	91.6	—	—	91.0	—
F-4 (3C) Waukesha	98.7	96.7	—	96.3	96.3	—	S 1.88
F-4 (3C) Stancal, % S/M	—	—	97.0	—	—	96.5	—
Modified F-4 at 125 F IAT, % S/M	—	—	91.6	—	—	91.2	—
Modified F-4 at 225 F IAT, % S/M	—	—	91.0	—	—	88.7	—
Modified F-4 at 300 F IAT, % S/M	—	—	88.2	—	—	87.8	—
Olefins, %	4	1	Trace	—	Trace	Trace	—
Aromatics, %	10	3	4	—	3	4	—
Paraffins, %	56	55	55	—	48	50	—
Naphthenes, %	30	41	40	—	49	45	—
Tel per U. S. gal, ml	—	3.98[b]	4.10	4.22[b]	4.28[b]	4.42	4.54[b]
Acid He at F	—	—	3	—	—	3	—
Bromine No.	6.8	1.9	0.3	—	0.4	0.3	—
Gravity	64.1	65.3	64.3	61.4[b]	61.7	61.5	61.1[b]

All inspection data by Standard Oil Co. of Calif. unless otherwise noted.
[a] These samples taken two weeks after tests on fuels A and D, respectively, due to desire for larger samples for further analyses.
[b] These inspections by American Airlines' laboratory.

Table 5 – Fuel Inspection Data[a] – Flight Volatility Tests
New York and Boston, February, March, 1941

Fuel	AAL	V-6	100	AAL[b]	V-6[b]	V-4A[b]
Sample No.	1	2	5	X5-41	X4a-41	X6-41
Location	New York	New York	New York	Boston	Boston	New York
Taken from	R. A. Tank	R. M. Tank	Truck hose	L. M. Tank	R. A. Tank	Drum
Date Taken	2-15-41	2-16-41	2-18-41	3-18-41	3-18-41	3-21-41
ASTM Distillation:						
Initial	116	122	110	112	111	108
10%	157	172	150	158	175	172
50%	200	232	208	200	225	234
90%	243	276	254	240	276	284
96%	256	301	274	268	317	340
End point	285	325	290	286	324	350
Knock Value, IC	93.5	97.0	Iso + 0.01	—	—	—
Tel per U. S. gal. ml	3.64	3.22	2.86	—	—	—
Reid Vapor Pressure, psi	5.8	4.5	6.5	—	—	—
Gravity, deg API	63.9	64.5	68.0	64.7	65.6	66.7
Corrosion	Negligible	Negligible	Negligible	Negligible	Negligible	Negligible

[a] Samples 1, 2, and 4: Shell Oil Co.'s Wood River research laboratory. Samples X5-41, X4a-41, and X6-41: American Airlines' laboratory, New York.
[b] Cold starting tests only, not used in flight tests.

altitude with a full cold carburetor air temperature of approximately 60 F. The particular point to which attention should be directed is that at a fuel/air ratio of 0.072, there is a power difference of approximately 80 hp or 13% between fuels *A* and *D*. Everyone should agree that, since the knock value difference between these two fuels is only 0.7, hardly beyond the accuracy of the F-3 test method, over 10% of this reduction in power must be due to other factors. The only other primary difference noted from the inspection data of the two fuels is the volatility difference of 40 F for the 90% distillation point. It is our contention that the greater portion of this 13% reduction in power is due to poor distribution of the less volatile fuel. Note that the more volatile fuel meets desired commercial limits, whereas the less volatile fuel only fulfills the 90% distillation limit in Army-Navy Specification No. AN-F-26 for 91/96-grade fuel.

Since, during winter operations, 20% of the flying may be done at a carburetor air temperature of 100 F and 5% at this temperature even during summer operations, a set of data was obtained similar to the previous tests except with the c.a.t. set at 100 F. Fig. 3 illustrates the results obtained. Here again an indicated power loss of 19% is to be observed at 0.072 *F/A* when a fuel with a supposedly satisfactory 90% distillation temperature is substituted for one with the preferred airlines' limit of 257 F. In order to have suitable knock-free performance at same power with the less volatile fuel, we had to go from a mixture ratio of 0.072 to one of 0.076. In doing this, it means we are essentially adding 1¢ per gal to our fuel cost. This is definitely prohibitive not only at the present time but, as pointed out in estimated future airline consumption figures, it would mean a $5,000,000 annual loss to the air transport industry in 1954.

Fig. 4 refers to tests again made under the same basic operating conditions except the c.a.t. has been raised to 150 F as a means of gathering further data. At this temperature, the two fuels behave in about the same manner at a mixture ratio of 0.076. This, we believe, is further proof that the volatility characteristics, as judged by distillation limits, are of major importance as related to power losses at the same fuel/air ratio or increased specific fuel consumption at same power. This is due, we believe, to poor distribution with attendant lower knock-limited performance as noted when a less volatile fuel is substituted for one of satisfactory higher volatility.

One question which was raised in connection with the two 91-octane fuels used was that concerning their temperature sensitivity. In order to answer this, the data given in Table 4 are pertinent. The F-4 (3C) and modified F-4 knock ratings, as determined by the Standard Oil Co. of Calif., show no marked difference with respect to this property for the two fuels studied.

Back in 1941 when the first concern of the petroleum people relative to increasing the permissible spread in distillation limits began to manifest itself, American Airlines, Inc., United Air Lines, Inc., and Pan American Airways, Inc., ran the first real flight evaluation tests ever conducted on fuels of differing volatility characteristics. In these tests, as in most current tests being carried on by the military services, Phillips Petroleum Co., and TWA, the knock-values of the fuels were purposely chosen to be sufficiently high in order to eliminate any possibility of detonation interfering with the results which might be observed.

In consequence thereof, no results similar to the 1944 tests were then observed, although the fuels as listed in Table 5 were essentially identical with respect to volatility differences. The only noticeable difference was a decrease in power of 19 hp or 1.83% for the less volatile fuel (278 F, 90% point) under take-off conditions and 10 hp or 1.1% decrease under rated power conditions on single-engine performance.

The connotation from a comparison of the above fuel flight studies is that actual operating conditions must always be observed for any given fuel, engine, and aircraft combination. For instance, from a theoretical and pure research standpoint, preliminary determination in flight tests of changes in fuel volatility characteristics without interference from adverse detonation is sound, but, from the practical viewpoint of the user, such investigations must be brought down to reality and, in the case presented, knock-limited performance was the restricting factor. Flight volatility tests per se must then take into consideration knock-limited performance, if their results are to be of ultimate value to the airlines.

I personally do not see how the two fuel quality requirements, detonation and volatility, which are of most concern to the air transport industry can be investigated separately. These two characteristics are too closely interrelated.

■ Summary

In order to improve safety, economy, and quality of aircraft-engine fuels, it is of the utmost importance that complete understanding and agreement be maintained between the fuel refiner, engine and airframe manufacturers, and the air transport operator. With this in mind, there

are many committees continually and actively working on a cooperative basis. It is their job to obtain and coordinate all the latest technological developments related to aviation fuel problems and to act as a guide in directing further research and development tests in a manner to hasten such work toward its final goal; as well as to prevent too great a duplication of similar tests by various companies independently.

Most cooperative work relative to aviation fuels is now conducted under the guiding hand of the Coordinating Research Council through the Coordinating Fuel Research (CFR) Committee's Aviation Fuel Division (AFD). There are various projects and small working groups or panels continually investigating and analyzing all technical tests and aspects of aviation fuel problems along the lines of volatility, detonation, satisfactory additives, vapor pressure characteristics, and the like. Naturally, during these times, close liaison is maintained with the military services and, in fact, most of the work is currently being done for them and is hence considered as restricted or confidential. Be assured, however, that all the mass of data continually being assembled will, in the long run, be of immense benefit to the aircraft industry. Reference may be made to a paper by Veal.[2]

A few of the typical specific problems yet to be answered and pertinent to the subject may well be summarized at this time. Detailed data cannot, of course, be revealed.

Knock rating procedures are actively being investigated. With the advent of many new high-octane blending agents and methods of refining which result in fuels of differing hydrocarbon compositions, test methods must, of course, be revised to prevent acceptance of unsatisfactory fuels and yet not be unduly restrictive. A method of rating fuels under the complete range of fuel/air mixtures found in service is one of the problems requiring an answer; or at least a method which will give comparative mixture response data as well as information relative to temperature sensitivity characteristics.

Volatility, of course, as previously discussed, is a perennial and prolific source of trouble as regards desire for information; particularly so since it is so closely tied in with vapor pressure properties of fuels. The principal reason for this is due to the fact that almost all specification requirements which concern this property must be a compromise between satisfactory engine operation and the economics of petroleum refining. It seems that almost all items relative to this fuel property are restrictive; still, safety in flight must be maintained and flight operations must be economical. Most current tests, therefore, are being made with the hope that the widest permissible distillation range will be found which will not penalize the air transport operator as regards consumption, safety, or other related difficulties. The following requirements are being considered: maximum and minimum values for the 10% distillation point, maximum value for the 90% and final boiling points, and vapor pressure limits. Another factor which must be given a great deal of future study is the detonation-limited characteristics of fuels with varying volatility characteristics in combination with different full-scale engines operated under varying flight conditions.

There are many more miscellaneous and some apparently small items concerning which clarification and interpretation are needed; for instance, improved gum and stability tests which can readily be related to service and storage experience, effect of various sulfur limits on corrosion of engines, the question of viscosity and/or gravity versus carburetor metering and compensation characteristics, satisfactory color identification of aviation fuels of different octane grades, and dyes for same, which will leave less deposits, and gum inhibitors of improved type or preferably improved stability of fuels as regards gum formation and lead deposition without the use of inhibitors.

[2] See SAE Transactions, Vol. 53, July, 1945, pp. 387-391: "Review of Coordinating Research Council Activities," by C. B. Veal.

■ Future Grades

Everyone has been exposed to the publicity programs continually being broadcast by the fuel vendors concerning their new "super fuels." Sentiments of this sort are akin to Nietzche's philosophy of glorifying the superman or, for that matter, to the weekly exploits of "Superman." Frankly, the airlines would prefer less ballyhoo and a more down-to-earth, realistic approach to the development of economical and good quality fuels. We, assuredly, are not in the market for "super fuels" at premium prices.

You may well ask what grade fuels the operators expect to purchase. Naturally, this could only be answered accurately if we knew what type engines and aircraft would be available at any given time. My own interpretation of the trend is that the bulk of aircraft-engine fuels will, for five to ten years, be grades 91/96 and 100/130 with some possible demand, to cover special operating conditions for a grade higher, say one similar to the prewar iso-octane plus 0.8 ml tel per U. S. gal.

The only basis on which the author makes this assertion is based on discussion with members of the engineering profession whose job it is to decide on future equipment. Certainly I am not qualified to discuss the highly controversial details of aerodynamic calculations, flying characteristics, and the like. Suffice to say that practically all projected postwar 4-engine aircraft will be landing-weight and space-load limited.

Typical examples which might be mentioned are two types of aircraft with respective take-off weights of 93,000 and 100,000 lb and landing weight limits of 70,000 and 77,000 lb. These aircraft are capable of flying nonstop between London and Newfoundland (2400 miles) with adequate fuel reserve (400 miles + 2 hr of flying) under the worst weather to be expected, at maximum cruise power and with space-capacity payloads of 11,500 and 13,500 lb, respectively. The same generalizations apply to larger aircraft capable of flying nonstop New York to London.

From the above, it will be observed that the spread between the landing and take-off weight limitations permit a large fuel load, approximately 23,000 lb (3850 gal). In no instance is it indicated that such airplanes would suffer a payload penalty with even a 10% increase in fuel consumption. Since a 10% decrease under such operating conditions would mean little beyond the direct fuel saving, such a decrease could only justify a proportionate price increase, approximately a penny a gallon on a fuel with a base price of 10¢. However, the reverse assumption is logical, that is, since a 10% increase in fuel consumption would not penalize payload, a lower, hence cheaper, grade fuel could well be substituted particularly for cruising flight. This reduction in grade, rather than "super fuels,"

concluded on page 152

PERFORMANCE TESTING of WHEEL-BEARING LUBRICANTS

(Report on Grease Projects—Coordinating Lubricants Research Committee of Coordinating Research Council, Inc.)

by E. W. ADAMS*
Panel Leader
CLR Grease Projects

THIS paper covers cooperative work conducted by the Coordinating Research Council, through its Program and Analysis Group Working on Grease Projects of the CLR General Division, in response to an urgent request by the U. S. Army Ordnance Department. It so happened that I was appointed chairman of the special panel for this project, designated "CLLG-25-43, Recommendations Relative to Performance Tests on Wheel-Bearing Lubricants." This panel was organized with representation from the following laboratories:

California Research Corp.
Gulf Research & Development Co.
Kendall Refining Co.
Pure Oil Co.
Quaker State Oil Refining Corp.
Rock Island Arsenal, Ill.
Sinclair Refining Co.
Socony-Vacuum Oil Co., Inc.
Standard Oil Development Co.
Standard Oil Co. (Ind.)
Texas Co.
Tide Water Associated Oil Co.

[This paper was presented at the SAE National Fuels & Lubricants Meeting, Tulsa, Okla., Nov. 7, 1945.]

* Research Department, Standard Oil Co. (Ind.)

PRESENTED here is a description of the apparatus and procedure developed for the performance testing of wheel-bearing lubricants.

This work was conducted by the Program and Analysis Group Working on Grease Projects of the CLR General Division of CRC, in response to a request by the U. S. Army Ordnance Department. Dr. Adams was appointed chairman of the special panel for this project.

THE AUTHOR: E. W. ADAMS has both taught and practiced engineering, the former in the early part of his career. Receiving his education at the University of Illinois, Iowa State College, and the University of Wisconsin, the author was professor of chemistry at Kansas State Teacher's College. Since 1925 he has served on the industrial research staff of Standard Oil Co. (Ind.), where he is now section leader of research in charge of development of lubicating greases, industrial lubricants and insecticides.

I take pleasure in presenting the results of the united efforts of this group, which has so diligently functioned under the auspices of the CRC.

■ Need for Performance Test

The need for a performance test upon U. S. Army Specification 2-108 Greases was first recognized when it was noted that certain greases manufactured in other countries and which met the specifications were, in certain cases, unsatisfactory in service. Subsequent observations in the United States also indicated that greases could meet the subject specification in its entirety and yet prove inferior in wheel bearing application. The following are typical of observations received from the field:

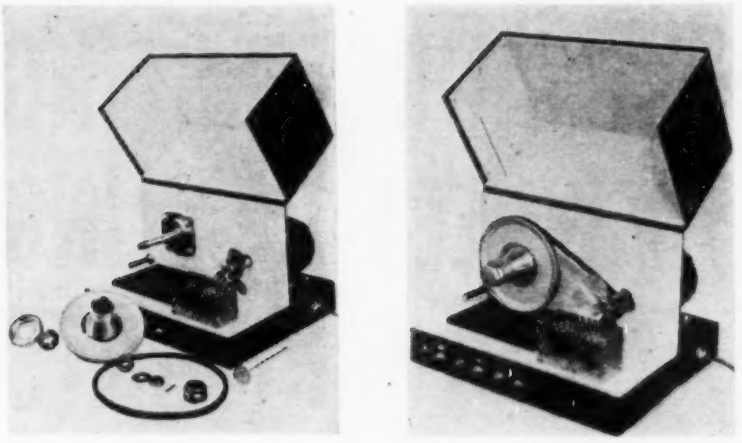
1. Apparently, the adhesive characteristics of this grease are not what they should be since a bearing carefully and thoroughly cleaned and dried and as carefully and thoroughly packed and adjusted is practically greaseless after a few revolutions of the wheel. The grease unwinds itself out of the spaces between the bearing and into the axle housing just as though it were a piece of yarn being pulled out.

2. After a short operation of the vehicle the bearings appear to be dry and no adherence of the grease, either to the rollers or to the races, is noticeable. The grease appears to knit and adhere perfectly to itself but departs from the bearings and it seems to work itself out from the bearings and fills up the hub cavity.

3. Mortalities of wheel bearings have become exaggerated beyond reasonable expectancies and are not altogether due to the type of service which they are receiving. Inner and outer races are pitting and the ground surface flaking at the point of pitting. Close observance at intervals of 100 to 1000 miles discloses that the lubricant does not have adhesive qualities necessary to lubricate the bearing surfaces. After short intervals of operation the bearing races and rollers are dry.

■ Program of Panel

It was recognized by the panel that various apparatus, more or less complicated, had long been in use by the



■ Fig. 1 – Test machine – consists of front-wheel hub fitted with 1933 Ford races and bearings, is driven at 430 rpm (equivalent to 40 mph) by means of V-belt and motor encased in insulated removable hood.

manufacturers of wheel bearing greases for evaluation of their respective products. It seemed desirable, in this case, to select a simple and relatively inexpensive type of test machine that could be made available to all suppliers and that would give a reliable indication of service characteristics, applicable to greases covered by the current items of U. S. Army Specification 2-108, as shown below:

1. Tendency of grease as such, or of separated oil, to leak from the wheel bearing in service.
2. Ability of the hot grease to hold its form in the hub.
3. Tendency of grease toward abnormal consistency changes in drastic service.
4. Tendency of grease to separate oil or soap in drastic service.
5. Tendency of grease to fail to lubricate bearings, that is, to allow balls or rollers to run dry, although adequate grease remains in the hub.
6. Tendency of grease to form deposits upon the bearings or races.

With the above in mind, a detailed study was made of laboratory assemblies of both Ford and Chevrolet front wheels and bearings in setups typifying various wheel-bearing lubricant testers already extant and upon which appreciable background was available. Greases of good and bad service behavior were employed as a basis for evaluation of the test mechanisms. As a result of this study, a hub of special design was developed.

■ Description of Apparatus

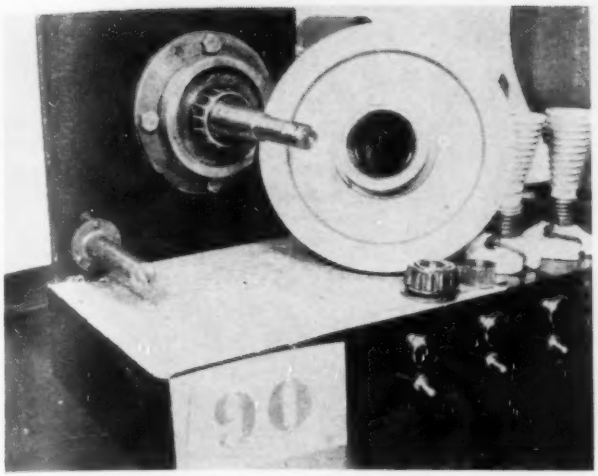
The tester (Fig. 1), consisting of a front-wheel hub fitted with 1933 Ford races and bearings, is driven at 430 rpm (equivalent to 40 mph) by means of a V-belt and motor encased in an insulated, removable hood. Means for thermostatically controlling the temperature have been built into the base of the device to make it completely self-contained. A fan on the motor shaft circulates air over the heaters and hub. The regular grease retainer is not used and in its place is a ring for catching any grease which leaks from the inner end of the hub. This is removable and provides a method of determining the grease loss. This grease-catching ring slips over the spindle and is held in place by the large bearing. The regular grease retainer is not used for two reasons: (1) it is desirable in a laboratory test to accelerate any leakage which may occur, and (2) defective retainers are found in actual service.

It was originally believed the machine could either be satisfactorily fabricated by individual shops or purchased from a supplier. However, experience gained in correlative tests employing commercial hubs of several makes, several

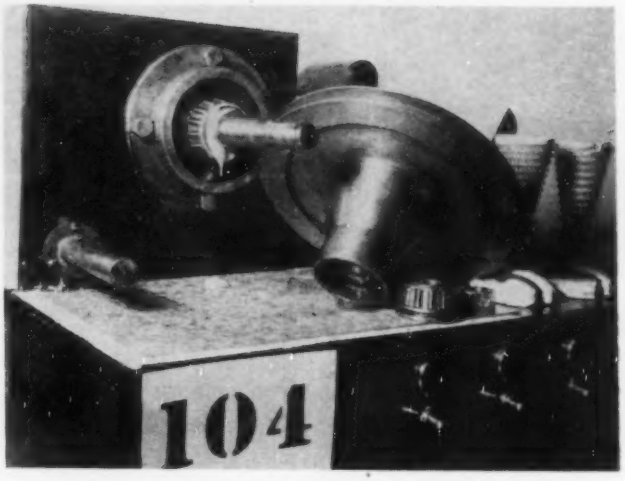
Table 1 – Original Data on Proposed Tester

CLLG No.	Appearance	Lubrication	Deposit	Leakage, %	Flowage from Hub	Flowage from Spindle	Remarks	Rating	Service
90	Green – smooth fiber	O.K.	None	None	Holds form	None on	—	Pass	Good
91	Yellow – fibrous	O.K.	None	0.5	Runs out – foamy [a]	Drops off	—	Fail	Bad
92	Brown – fibrous	O.K.	None	10 [a]	Runs out [a]	Some drops off	Difficult to pack	Fail	Very Dubious
93	Black – fibrous	Dubious	Slight [a]	21 [a]	Runs out [a]	Thin film – stays on	—	Fail	Very Dubious
94	Black – very fibrous	Very poor	Much [a]	29 [a]	Runs out [a]	Thin film – stays on	Difficult to pack – very noisy	Fail	Bad – failure
95	Black – grainy	Dubious	Slight [a]	None	Slumps out [a]	Some drops off	—	Fail	Unknown
96	Green – fibrous	O.K.	None	22 [a]	Runs out [a]	Some drops off	—	Fail	Unknown
97	Green – smooth	O.K.	None	None	Holds form	None	—	Pass	Good
98	Green – fibrous	O.K.	None	11 [a]	Slumps out [a]	Stays on	—	Fail	Bad
99	Black – fibrous	Dubious	Slight [a]	4.5 [a]	Runs out [a]	Drops off	—	Fail	Dubious
100	Brown – smooth	Dubious	Slight [a]	(5-front) [a]	Little in hub	Heavy, fibrous, stringy [a]	Gets very fibrous and heavy	Fail	Dubious
101	Brown – fibrous	Dubious	Slight [a]	Trace	Runs out [a]	Drops off	—	Fail	Dubious
102	Brown – fibrous	O.K.	None	17.3 [a]	Holds form	Stays on	—	Fail	Dubious
103	Green – fibrous	O.K.	None	None	Runs out [a]	Drops off	—	Fail	Good
104	Yellow – fibrous	O.K.	None	Trace	Free oil – slumps [a]	Small amount – drops off	Gels [a]	Fail	Bad
105	Amber – fibrous	O.K.	None	None	Holds form	None	Sets up Cold [a]	Pass (?)	—
106	Green – fibrous	O.K.	None	None	Runs out – foamy [a]	Considerable – some drops off	—	Fail	—
109	Green – smooth fiber	O.K.	None	None	Free oil – slumps	None	Gel [a]	Pass	Good

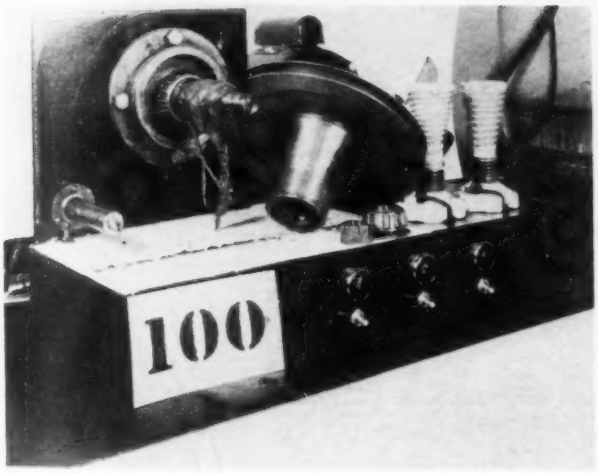
[a] Reasons for failure.



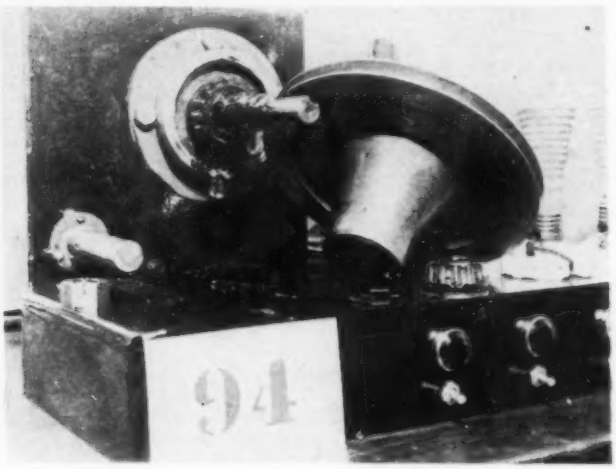
■ Fig. 2 – Example of perfect test



■ Fig. 4 – Example of separation of free oil – tendency to gel



■ Fig. 3 – Example of transposition from originally very smooth structure to very fibrous, ropy condition – slight deposit on bearings



■ Fig. 5 – Example of 29% leakage – extreme puddling, large deposit on bearings, noisy running

specially made hubs, and the first experimental model of the machine built by an apparatus supplier served to emphasize the necessity for detailed consideration to the hub and spindle assembly if test results were to be reproducible. Furthermore, these correlative tests demonstrated not only the need for care upon hub and spindle but also upon insulation, heat loss through the spindle support, and actual temperatures attained in the back bearing, as indicated by thermocouple measurements taken through the rear of the spindle, bored to a depth of 1½ in. (Spindles of all machines are, therefore, so drilled for this purpose.)

It is upon this basis, therefore, that recommendation was made in the interest of uniformity of results (repeatability and reproducibility) that all machines be purchased from one supplier.

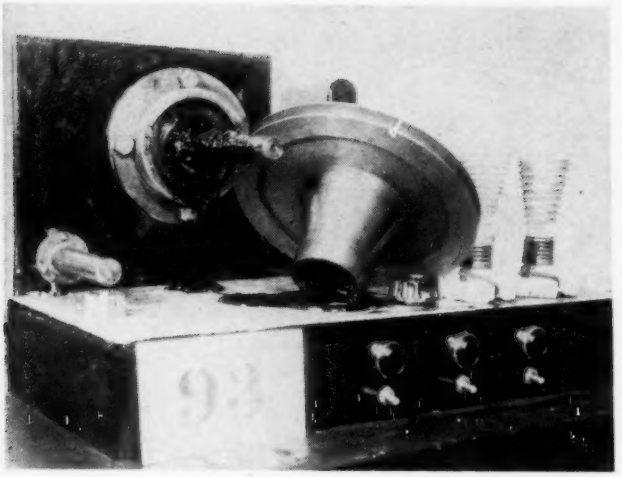
■ Test Procedure[1]

A total of 90 g of grease is used for each test. The bearings are carefully packed by hand with approximately 5 g of grease (about 2 and 3 g in the smaller and larger bearings, respectively) and the balance is placed in a uniform layer on the inside of the hub. A very thin film of grease is applied to the outer races contained in the hub. The 90 g of grease fills the hub practically even with the wheel races and, with the exception of very fibrous greases, can be readily and uniformly distributed by use of a small spatula.

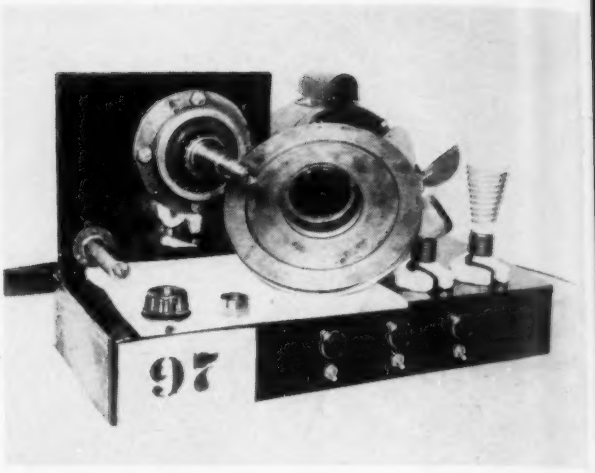
The leakage cup is placed in position and the large or inner bearing, packed with the grease being tested, is next placed in its proper position on the spindle. The wheel and outer or small bearing are then put on, followed by the loose fitting retainer washer and the castellated nut for holding the wheel assembly in place. The assembly is tightened until the wheel ceases to coast when spun by hand and the castellated nut is then slacked off until the wheel rotates quite freely without end play (approximately one-sixth of a turn). The hub is then locked in place by insertion of the cotter pin. After the hub cap is screwed on, the V-belt is finally put into place and the cover closed.

When the apparatus is thus assembled the motor and both heaters are turned on. Each test is run for 6 hr from the time the motor is started and the heat switched on. (In case of questionable results in a 6-hr run a second test

[1] The recommendations contained in this section are included in "Recommendations Relative to Performance Tests on Wheel-Bearing Lubricants" (CRC Designation L-24-745). The procedure is made available as research information only, and is not to be construed as a specification or standard, since the Coordinating Research Council, Inc., does not promulgate specifications or standards.



■ Fig. 6 – Example of 21% leakage – puddling, slight deposit on bearings



■ Fig. 7 – Example of perfect test

of 16-hr duration can be made, the same requirements being imposed upon the lubricant. The need for the longer period of test will be found in rare instances only.)

The temperature of the spindle is brought to and maintained at 220 F. This will be attained by an ambient temperature near the hub of approximately 235 F, varying somewhat with room temperature, air currents, and so on. This temperature should be reached in a maximum of 30 min, 20 min generally being adequate. An ambient of 235 F will be held with the continuous heater on and the intermittent cutting out at about 235 F.

■ Evaluation of Grease

The Ordnance Department now has under consideration as a revision to U. S. Army Specification 2-108A, the inclusion of the following criteria to be used for the evaluation of the grease at the end of the 6-hr running period:

1. The apparatus, immediately at the conclusion of the test, is dismantled and the wheel inclined upon the bench, resting upon the edge of the pulley and outer hub. Flowage of grease from the hub shall not, *per se*, be cause for rejection, so long as the material is still of grease consistency. However, abnormal loss of grease consistency to the extent of becoming quite thin or puddling shall be cause for rejection.

2. Abnormal change of grease structure from the original shall be cause for rejection. (This refers to either smooth or short fiber grease assuming a very fibrous ropy structure, setting up, gelatinizing, or forming a dry, brittle gel structure.)

3. Leakage in excess of 10 g (11%) of grease and/or oil into the grease retainer at the back of the hub shall be cause for rejection.

4. Any adherent deposit of varnish, gum, or material of lacquer-like nature upon the bearings or races which is in evidence after slushing for 2 min at room temperature in naphtha (as specified in ASTM Designation: D91-40) for removal of the grease film shall be cause for rejection.

Table 2 – Summary of All Data

	Proposed Testers			Other Machines			
	Final Model		Laboratory C Model				
Laboratory Sample	A	B	C	D	E	F	G
G-90	P	P	P	P	P	P	P
G-91	F	F	F	F	—	F	F
G-92	F	F	F	F	—	F	F
G-93	F	F	F	F	—	F	F
G-94	F	F	F	F	F	F	F
G-95	F	F	F[a]	F	—	F	P[a]
G-96	F	F	F	F	—	F	F
G-97	P	P	P	P	—	F[a]	P
G-98	F	F	P[a]	F	—	F	F
G-99	F	—	—	—	—	—	—
G-100	F	—	—	—	—	F	—
G-101	F	—	—	—	—	—	—
G-102	F	—	—	—	—	—	—
G-103	F	F	P[a]	F	F	P[a]	F
G-104	F	F	F	F	F	F	P[a]
G-105	P	—	—	—	—	—	—
G-106	F	—	—	—	—	—	—
G-109	P	—	—	—	—	—	—

P—Pass; F—Fail.

[a] Tests not agreeing with rating given greases upon final model of tester in A and B aboratories.

■ Lubricants Tested

A total of 18 lubricants of good or bad service behavior was studied, as listed below:

Sample	Source	Supplier's Information
G-90	Cooperator	Good
G-91	Cooperator	Bad
G-92	Ordnance	Very questionable
G-93	Ordnance	Very questionable
G-94	Ordnance	Bearing failures
G-95	Ordnance	Unknown
G-96	Ordnance	Unknown
G-97	Cooperator	Good
G-98	Cooperator	Bad
G-99	Ordnance	Questionable
G-100	Ordnance (foreign)	Questionable
G-101	Ordnance (foreign)	Questionable
G-102	Ordnance (foreign)	Questionable
G-103	Cooperator	Good
G-104	Cooperator	Bad
G-105	Cooperator	?
G-106	Cooperator	?
G-109	Cooperator	Good

■ Original Data

The original data obtained upon the 18 test greases are shown in Table 1, the items marked "a" indicating reasons

for failure, as judged by the criteria under consideration by the Ordnance Department. It will be noted that every grease of questionable or bad service history in the experience of the supplier failed in the test. In only one case was a grease reputed to be satisfactory in service rated as failing.

Photographs taken at the conclusion of certain of these tests are shown in Figs. 2-7, inclusive.

■ Cooperative Data

It appeared at this stage that the test under study was indeed indicative of service behavior. However, before proceeding further it was felt that these same greases should be tested in machines of other design long used by various members of the panel for evaluation of wheel-bearing lubricants. Table 2 shows a summary of the resulting data, the items marked "a" indicating ratings in disagreement with those secured upon the final model of the tester, in use at that time in two laboratories. While there are a few discrepant results it is believed this is partly due to divergent ideas as to what is required of a grease for satisfactory wheel-bearing lubrication.

■ Present Status

With this background an equipment manufacturer was authorized to build these machines. A total of 16 has been fabricated and delivered to as many laboratories. Another lot of 25 is practically completed.

Coordinating tests are now under way upon the original 16 machines and preliminary data look very uniform.

■ Conclusions

This test, as proposed, is fairly liberal. Many of us believe, in contrast to the 10 g allowable, that no leakage from the inner bearing should be tolerated. Some feel that migration of grease to the front of the hub is bad. Others would go further and consider unsatisfactory not only greases that puddle or flow as liquid from the hub upon dismantling, but also those that slump from the hub in a mass which is still of grease consistency. A few cooperators feel the test should be of longer duration than 6 hr. We, in our own laboratory, have, however, never found a grease that would not pass the criteria now under consideration by the Ordnance Department at the end of a 16-hr test that would not also have been eliminated at the conclusion of a 6-hr period.

Final agreement upon details of procedure and interpretation of results as noted above must, we believe, await the accumulation of greater experience with this machine upon the part of industry through cooperative effort.

However, the test as at present agreed upon will in the meantime eliminate inferior wheel-bearing lubricants. If conscientiously employed by suppliers it is believed that the average quality of wheel-bearing greases will be markedly improved and that the headaches of both supplier and consumer due to poor field performance will be alleviated, if not obviated.

DISCUSSION

Discusses Application Of Test to Army Specs

—MAJOR N. W. FAUST

U. S. Army Ordnance Department

GREASES used for wheel-bearing lubrication of Army equipment are procured under U. S. Army Specification No. 2-108, Grease, General Purpose No. 2. This specification reduced to its barest essentials covers a sodium-soap grease containing approximately 15% soap and a heavy mineral oil; in other words, the ordinary commercial wheel-bearing grease. Experienced manufacturers know there are a number of ways to put these two ingredients together, but not all of these ways produce a good wheel-bearing grease. With the ever-increasing volume of greases demanded, it was inevitable that greases of questionable quality would find their way into the Army supply channel because composition does not permit a sharp line of demarcation between good and poor service performance.

One point which should be emphasized and for which Dr. Adams and his coworkers should be complimented is the simplicity of the equipment and the ease of operating it. Many of the refinements dear to the heart of the investigator have been omitted in order to provide an instrument suitable for the primary purpose; namely, quality control for the procurement of wheel-bearing greases.

The Ordnance Department is now taking steps to incorporate the test method and suitable limits in U. S. Army Specification No. 2-108. It may appear paradoxical to include this test and yet retain the composition features of the specification. But this serves to emphasize another point which should not be overlooked and that is the danger of extending the use of the test beyond its intended scope or of over-interpretation of the results. In its simplest form, this test indicates whether leakage, changes in texture, consistency, or oxidation will occur. But other factors enter into wheel-bearing grease performance, such as the viscosity characteristics of the mineral oil, the consistency, the soap content, and type of soap. Those factors could be evaluated by one or more, probably several performance tests, but the equipment, personnel, and time involved are in the category of research rather than routine inspection. Fortunately, however, extended commercial and military experience have fairly well defined the limits for those factors in terms of composition and physical characteristics and thus the necessity for incorporating extensive performance tests disappears.

As mentioned previously, the performance test is being incorporated into U. S. Army Specification 2-108 and it may be of interest to discuss this phase a little more fully. Referring again to the four characteristics evaluated by the instrument; namely, leakage, changes in texture, consistency and oxidation, it is obvious that evidence of leakage or oxidation is based on actual measurement or observation, whereas abnormal changes in texture or consistency are matters of interpretation. The average person will experience no difficulty with the former two evaluations, since the leakage will be either above or below 11% and deposits will either be present or absent. The latter two evaluations require considerably more experience than the average person has had in order to decide whether a change in texture or consistency is abnormal or normal.

Obviously, the use of the performance test as outlined for procurement inspection would lead to considerable difficulty in obtaining adequate supplies, but on the other hand, the use of the test for qualification only would not guarantee adequate quality because there would be no assurance that subsequent manufacture would duplicate the qualification sample. Both of these factors will be taken into account, however, by requiring the performance test for qualification and a modified performance test for inspection. The former includes evaluation of all the characteristics outlined by Dr. Adams, whereas the latter will include only determination of leakage and evidence of oxidation. Admittedly, this is not a perfect solution, but it will ensure that the manufacturer can produce initially a satisfactory wheel-bearing grease and ensures, in so far as possible, that he maintains this quality in subsequent production but avoids procurement problems by eliminating questions of interpretation.

Dr. Adams concludes his paper by pointing out that the final form of the test method and interpretation of results will be established after more experience has been gained with the equipment, but the use of the test in its present form will reduce the number if not eliminate entirely greases of poor service characteristics. The Ordnance Department shares this view, but intends to continue in this cooperative work on the finalization of a performance test for wheel-bearing lubricants.

Factors Pertaining to Ins
AIRCOOLED AI

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and
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CONSIDERATIONS of aircraft-engine installation singular to the inverted, in-line, aircooled type of engine are presented here in the light of current engineering development in this field. Items of this installation technique discussed are: cooling aspects, cowling arrangements, oil cooling systems, induction systems, engine mounts, and exhaust systems.

Much of the information presented is based upon experience gained from wind tunnel and flight tests with Ranger engines.

The application of design considerations discussed under the cooling aspects of the aircooled engine has been proved satisfactory in flight test operations.

Latest methods of treatment of these factors are described. Substantial advancement has

THIS paper discusses those factors involved in the installation of the inverted, in-line type of aircooled aircraft engine. The material is developed to emphasize considerations singular to this particular type of engine. Discussion of existing Ranger-powered installations is included to illustrate their application. Extensive treatment of the large number of installation aspects common to all aircooled engine types is beyond the scope of this paper.

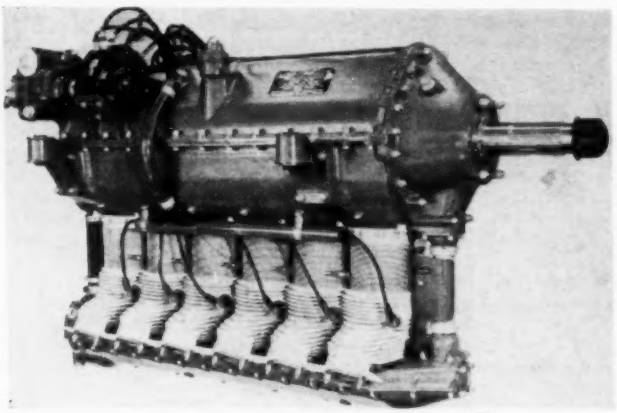
The information presented is based primarily upon experience gained from wind tunnel and flight tests with Ranger engines. Two of these engines, which have been under development at Ranger for some 15 years, one of the 6-cyl, 6-440 series, and one of the 12-cyl, V-770 series, are illustrated to facilitate visualization of the design features mentioned in the discussion.

Fig. 1 illustrates one of the 6-cyl series, started about 1929 with a rating of 110 hp, which has since been increased to 200 bhp. These engines use the inverted engine counterpart of overhead valves. The cambox is suspended from the cylinders and houses the camshaft and rocker arms.

Fig. 2 illustrates a 12-cyl engine, started about 1930 with a rating of 290 hp, direct drive, unsupercharged, and now delivers 620 hp for take-off, equipped with geared propeller drive and supercharger. Detail construction closely parallels that of the 6-cyl series with the addition of reduction gearing and a mechanically driven supercharger.

The material is arranged under six major headings: cooling aspects, cowling arrangements, oil cooling systems, induction systems, engine mounts, and exhaust systems. The topics are presented with respect to both the scope of their application and their degree of interest to the industry. The section of this paper which discusses the cooling aspects of Ranger engines deals primarily with the 12-cyl series.

[This paper was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Calif., Nov. 1, 1945.]



■ Fig. 1 – Ranger 6-440C series engine

■ Cooling Aspects

Engine cooling, despite the vast amount of research work expended on it, remains one of the foremost problems confronting aircraft and engine manufacturers. This problem is the one which probably requires the greatest joint effort on the part of both.

The study of cooling may be divided into two principal phases. One is the study of heat flow within the engine and its component parts, including the effect of cylinder design, development of optimum finning, choice of materials, and baffling arrangements affording optimum dissipation to the cooling air stream. The other phase, more important to the installation engineer, deals with the flow of cooling air and its efficient utilization. Paramount in the treatment of the latter phase is pressure cooling.

The principle of pressure cooling involves an air pressure chamber or reservoir on the upstream side of the engine cylinders, and cylinder baffling controlling and directing the high-pressure air in motion to assure its efficient utilization. The remaining or downstream side of the cylinders is in a low-pressure zone toward which the high-pressure air will flow. The pressure in this area may be further reduced by use of cowl flaps; this results in a higher pressure differential and greater airflow.

Fig. 3A illustrates conventional, in-line, aircooled cylin-

Installation of Inverted, In-Line AIRCRAFT ENGINES

been made in handling the ever-present cooling problem common to all types of engines. This has been achieved by applying the design knowledge gained through considerable wind tunnel and flight research.

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der baffling. The baffles are designed to direct the air toward the cylinder on the downstream side. On the upstream side the baffles are sometimes omitted because adequate cooling is available. A new development described below has modified this somewhat.

Fig. 3B illustrates schematically the Ranger developed, turbulent-flow baffle, which completely boxes the cylinder. Emphasis is placed upon inducing local turbulence within the area confined by the baffle in the interest of "scrubbing" action to maintain maximum thermal potential between cooling fin and cooling air. Attention is directed to the small auxiliary vanes within the baffle installed to assist in promoting turbulence.

Engines thus baffled display reduced airflow requirements for equivalent cooling. This results in lower required cooling power, lower required exit flap angles, and smaller engine cooling-air entrances.

An additional development relating to cooling-fin construction has resulted in improving the heat transfer of a finned cylinder to the cooling air stream, a measure necessitated by the higher specific output resulting from recent advances in engines and fuels. The low conductivity of the steel-barrel cooling fins used in early designs presented a recognized bottleneck in the heat flow to the cooling air stream. A substitute material possessing high conductivity, light weight, and workability was sought. The major problem was that of bonding the material to the steel cylinder barrel. This is accomplished in what is now known as the Al-Fin process, whereby an aluminum muff, into which cooling fins are later machined, is chemically bonded to a cylindrical steel shell to form the cylinder barrel. This method is discussed in detail by Piry.[1]

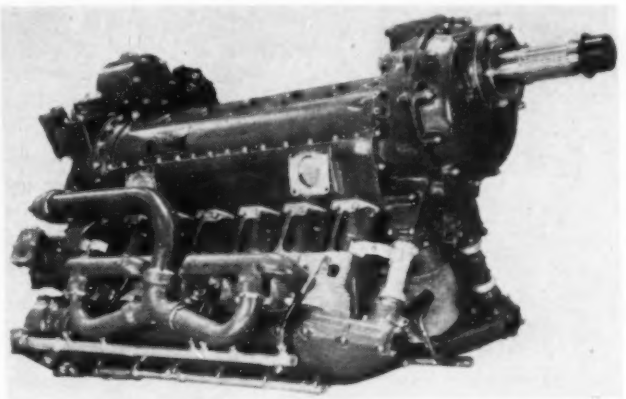
An aircooled engine derives its cooling airflow as a result of the forward velocity of the airplane, the fan effect of the propeller, and the pumping effect of the airflow past the cowl flaps. The airflow is affected by:

1. The entrance shape, its area and location.
2. Pressure chamber design.
3. Exit location, area, and shape.

Before discussing any particular installation with respect to the cooling problem, we shall review the fundamentals of the problem and their relation to the above three factors.

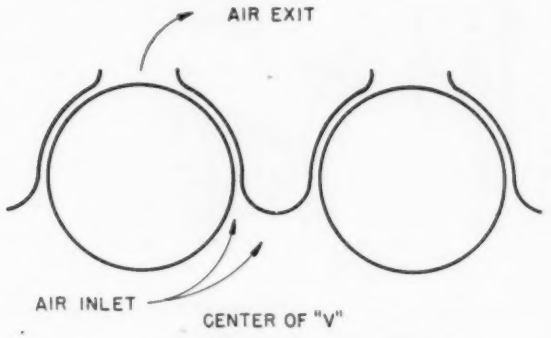
1. Entrance Shape, Area, and Location – In order to utilize the maximum fan effect of the propeller, it is evident that the centroid of the air inlet opening should be a substantial distance from the propeller hub. A triangular air entrance with the apex of the triangle toward the propeller hub is usually superior to a circular opening. In addition, such an arrangement minimizes propeller hub effects upon airflow into the pressure chamber. Wind tunnel tests substantiate this. They show that the triangular inlet gives better engine temperature-pressure distribution than does the circular opening. A dimensional check made on a radial to in-line engine conversion on the North American AT-6 airplane indicated that the inlet area centroid on the in-line installation was located at 48% propeller radius. The comparable point on the radial installation was located at 26.2% propeller radius.

To obtain equal cooling-air distribution to all cylinders, it is essential to provide nearly equal pressure over the full length of the engine. This can be realized if a reasonable portion of the velocity head of the free air stream is converted into static pressure head as the air enters the

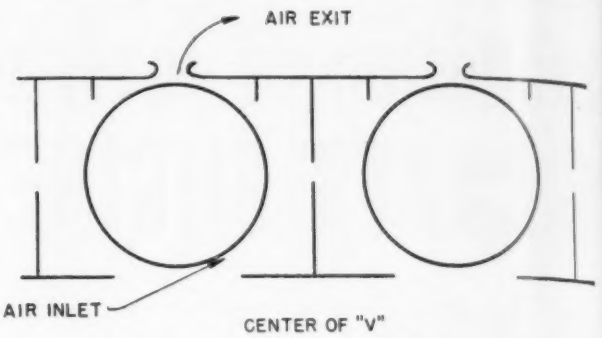


■ Fig. 2 – Ranger SGV-770C series engine

[1] See SAE Transactions, Vol. 53, November, 1945, pp. 630-639: "Cooling Characteristics of Steel- and Aluminum-Finned Cylinder Barrels for In-Line Aircooled Engines," by Marcel Piry.



■ Fig. 3A – Smooth-flow baffle



■ Fig. 3B – Turbulent-flow baffle

scoop. The static pressure at the front cylinders of the engine is primarily a function of the inlet velocity ratio, the ratio of the velocity in the entrance V_i to the free air stream velocity V_o and, secondly, a function of the efficiency of the inlet orifice. The effect of variation of this ratio upon the inlet air streamlines is pictured in Fig. 4. Tests have shown that inlet velocity ratios in the neighborhood of 0.4 are satisfactory. The usual procedure is to design the entrance with an inlet velocity ratio of 0.4 for high-speed level flight and to check this area for the best-climb condition. If the value of the ratio in the climb condition exceeds 0.8, a lower value, not lower than 0.3 is used for the level flight calculation. This proceeds until a satisfactory area is found. Both the high-speed and climb-speed calculations should be based on summer air for both sea level and the airplane critical altitude, and some reasonable compromise then made on the inlet area.

In the design of an efficient entrance, a smooth, well-faired lip with sufficient thickness to minimize the variation in pressure distribution with change in inlet-velocity ratio is necessary. Thin lips are undesirable because of their sensitivity to changes in angle of attack.

It is believed, and has been substantiated by practice, that a common entrance for cylinder cooling air, oil cooling air, and charge air gives excellent pressure recovery and pressure distribution over the cylinders, eliminates complicated and heavy ducting, and makes for ready accessibility to the engine for servicing and maintenance.

2. Pressure Chamber Design – A satisfactory method of cooling the inverted V type, in-line engine is illustrated schematically in Fig. 5. It consists of a high-pressure chamber in the V of the engine between the banks of cylinders. Air introduced at the scoop entrance is slowed down by diffusion after entering the pressure box and is largely converted into static pressure. A low-pressure chamber created by action of the cowl flaps is located on the opposite side of the cylinders.

Concepts as to the best means of baffling the cowl interior to form the pressure box have changed in recent years as experience grew. While early designs were not unsatisfactory, it has been found that better recovery resulted from the largest practical pressure box, as illustrated in Fig. 6.

3. Exit Location, Area, and Shape – The optimum air exit location has been determined experimentally from wind tunnel investigations conducted on one-fifth scale and full-scale models at M.I.T. as a Ranger project. A survey of static surface pressures, illustrated in Figs. 7 and 8 by the method of perspective contouring, indicates the pressure pattern for both the high- and the low-wing positions. These runs were made with no airflow through the cowl and flaps closed. Angle of attack was 9 deg. Negative pressures are shown by solid lines and positive pressures by broken lines. Similar tests with flaps opened and air flowing showed very slight changes in the pattern over that of the no-flow survey. These surveys indicate the following:

For low-wing airplanes:

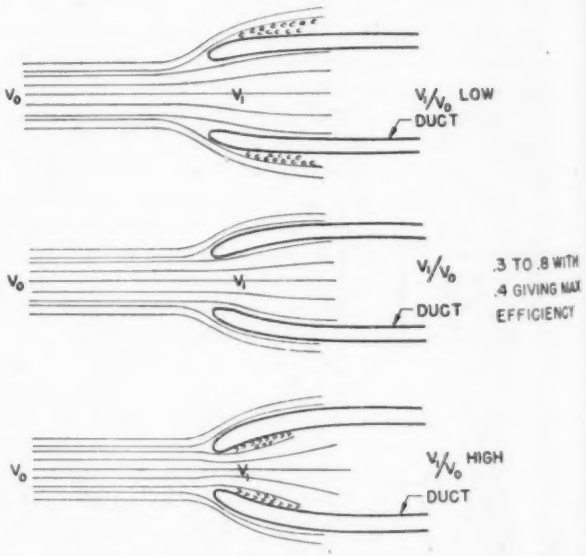
a. For maximum pressure drop Δp, locate exit at rear of cowl just aft of rear cylinder.

b. For maximum pumping efficiency η, defined as the ratio of the cooling drag coefficient resulting from internal losses C_{Dc} to the total drag coefficient resulting from change from no-flow to flow conditions over the cowl C_{Dp}, locate exit approximately at the fore-and-aft center of the cowl.

For high-wing or mid-wing airplanes:

a. For maximum pressure drop, locate exit at approximately the fore-and-aft center of the cowl.

b. For maximum pumping efficiency, locate exit at rear of cowl.



■ Fig. 4 – Effect of inlet velocity ratio on inlet air streamlines

In order to obtain adequate cooling in climbing-flight and ground operation, the exit area should be adjustable. This is necessary in order to reduce to a minimum the internal losses of overcooling, and the external drag at high velocities. The drag incurred by opening the flap greater than approximately 20 deg becomes relatively high with little improvement in flap action. If sufficient cooling can be provided for the critical conditions of flight with the flap openings not exceeding approximately 20 deg, then fulfillment of the less difficult requirements of level flight can be expected with the same flap in the faired position. Similarly, exit areas can then be selected to give just sufficient airflow across the engine for these minimum cooling requirements.

Power off, wind tunnel tests of various flap shapes (Fig. 9) indicated high specific pumping efficiency for a flap with a relatively long chord and proportionately narrow span, as shown in *b*.

Required cooling airflow is customarily expressed in terms of the pressure drop necessary to obtain flow. As a result of the wide variation in methods employed in measuring these pressure drops, flow calibrations cannot, in general, be compared satisfactorily.

Table 1 – Airplane Characteristics

Airplane	Type of SGV-770 Engine	Normal Rated Power	Bhp Take-off	Maximum Speed, mph	Ias Best Climb, mph	Engine Critical Altitude, ft	Total Inlet Area, sq in.
SO3C	C-1A	450	520	...	85	12,000	269
	C-2	500	550	159	...	8,000	269
XAT-14	C-1A	450	520	220	115	12,000	250*
AT-21	C-1	450	520	223	115	12,000	195*
	C-1B	450	520	223	115	12,000	...
XP-77	C-1B	450	520	350	...	12,000	192*
XAT-6E	D-4	465	[575	229	115	22,000	300*

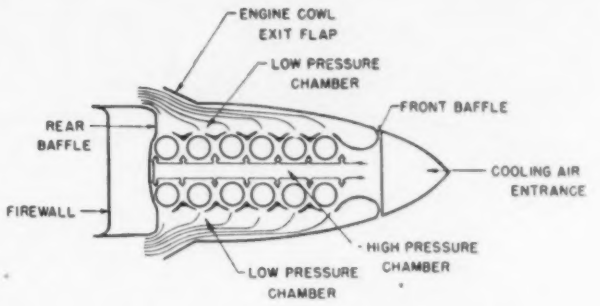
* Includes oil cooling and carburetor air.

It is likewise difficult to compare cylinder temperatures (the engine cooling index) in various installations without application of innumerable long and tedious corrections, necessitated by the influence of fuel flows, air speeds, cowling design, propeller characteristics, location of inlet in relation to the propeller, pressure chamber design, location of exit flaps, type of engine baffles, variations in cooling fins, and exhaust manifold design. The question of whether or not an installation cools is relatively easy to answer, but the question "why" presents a different problem.

Typical Installations – In order to illustrate clearly the general aspects pertinent to in-line, aircooled engine installations and to explain the treatment that has been applied to some of the detailed aspects involved, information on typical installations is presented.

The specific installations discussed are the Curtiss SO3C, the Fairchild XAT-14, the Fairchild AT-21, the Bell XP-77, and the North American XAT-6E, all using V-770 engines. All but one of these installations are shown in Fig. 10 through 13. Table 1 presents a few particulars regarding each installation.

SO3C – The nose cowl of an early V-770 powered airplane with a critical altitude of 12,000 ft was designed with a total air inlet area of 269 sq in. Although cooling was satisfactory, subsequent full-scale wind tunnel tests on the installation with windmilling propeller showed that satisfactory cooling could be achieved with an entrance area of 144 sq in. This was confirmed by flight test. The 125 sq in. reduction in entrance area showed little effect



■ Fig. 5 – Method of cooling Ranger SGV-770 series engine

on either the available pressure drop across the cowl, measured inlet to exit, or the pressure distribution along the cylinders.

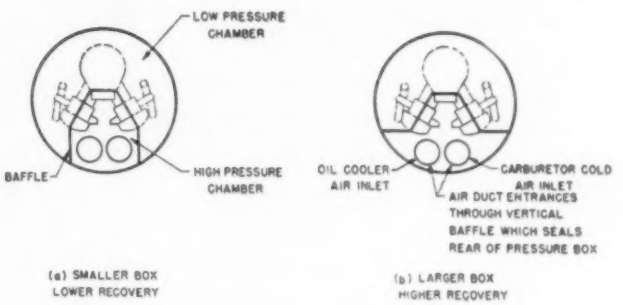
Fig. 14 shows the entrance and exit velocities obtained during power off wind tunnel tests with four different cowl openings. It will be noticed that even with the smallest entrance *D*, the entrance velocity is approximately 30% of that of the free air stream, while with a full cowl opening *A*, it is approximately 15%. This latter figure is low; the range of entrance velocities from 30% to 50% free air stream velocity is considered desirable.

The total pressure distribution was good both at the cowl entrance and fore and aft through the high-pressure chamber. The value at the entrance plane was 1.0*q*; this diminished gradually toward the rear of the V to a value of 0.85*q*.

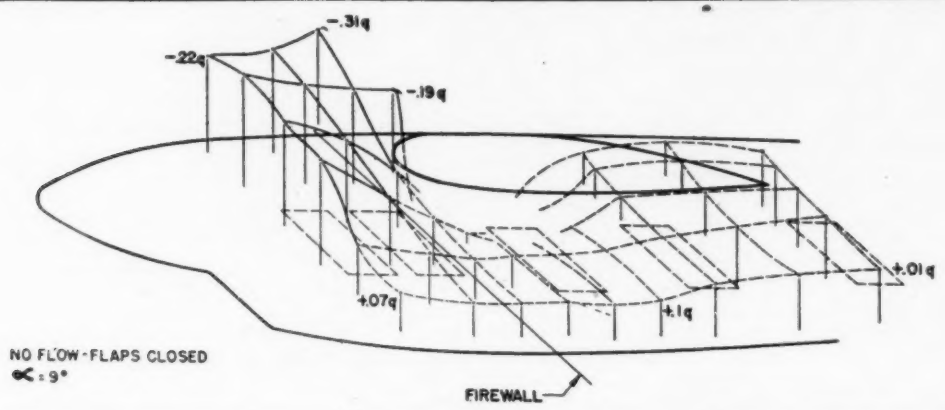
Figs. 15 and 16 show curves plotted from rated power flight test data on cylinder temperature variation with altitude for both turbulent- and smooth-flow baffles.

Further marked gains were achieved in cooling with the installation of the SGV-770C-2 engine equipped with aluminum-finned cylinders. This is illustrated by comparison of results of flight tests with the 770C-1 engine with steel barrel fins, and the 770C-2 engine with aluminum fins. Cylinder temperatures are presented in Figs. 17 and 18. Both engines used turbulent-flow baffles. The higher base temperatures experienced on the SGV-770C-2 engine are attributed to the increased normal rated power, raised from 450 on the 770C-1 to 500 bhp on the 770C-2.

XAT-14 – The XAT-14 was the forerunner of the AT-21. The air inlet system (Fig. 19) consisted of three


LOW PRESSURE CHAMBER
BAFFLE
HIGH PRESSURE CHAMBER
OIL COOLER AIR INLET
CARBURETOR COLD AIR INLET
AIR DUCT ENTRANCES THROUGH VERTICAL BAFFLE WHICH SEALS REAR OF PRESSURE BOX
(a) SMALLER BOX LOWER RECOVERY
(b) LARGER BOX HIGHER RECOVERY

■ Fig. 6 – Pressure box area plotted against pressure recovery in SGV-770 cowles



■ Fig. 7 – Static pressure distribution over side of nacelle – high wing

main ducts, the center duct for engine cooling air, the left-hand side inlet duct for oil cooling air; and the right-hand side inlet duct for the carburetor air. The high-pressure chamber was bounded within the V by the ducts on either side and by a vertical baffle at the rear. The low-pressure area was bounded by the cowl and included the accessory compartment.

The XAT-14 installation had eight adjustable cowl flaps arranged three on each side and two on the bottom center of the engine cowling, as shown in Fig. 20. The maximum flap setting was 45 deg, while at 0 deg the flap contour faired into the nacelle.

It will be seen from the plots of the corrected temperatures (Fig. 20) that the cylinder-head temperatures fell below the allowable, while the base temperatures were marginal on cylinder Nos. 2 and 6 of the left bank.

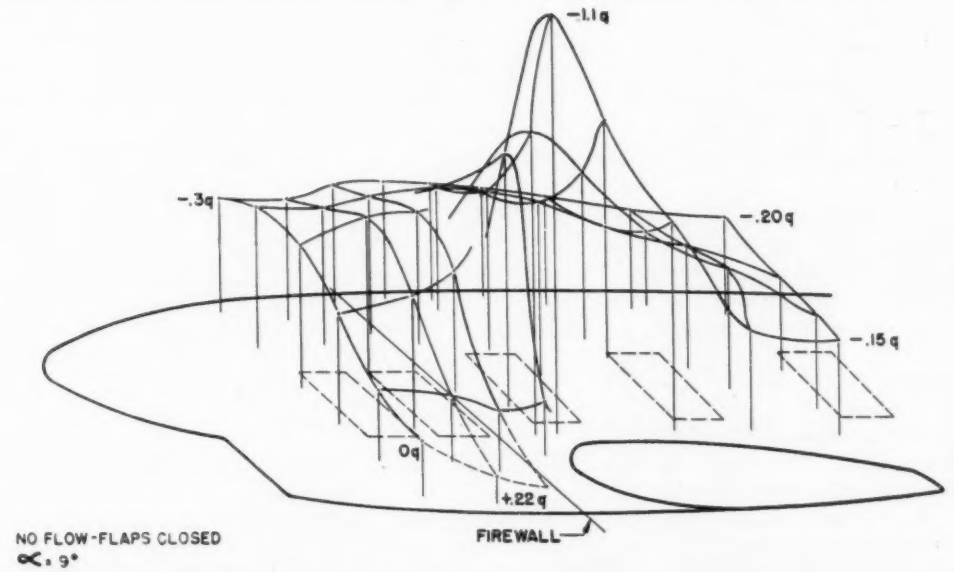
AT-21 – Both the SGV-770C-1C and SGV-770C-1B engines, which differed in carburetion, were used in the AT-21 airplane. The installations differed in the method of ducting air to the carburetor. In the SGV-770C-1B installation, cooling air was taken in through a common entrance (Fig. 21) and directed into the V or high-pressure chamber from which the engine cooling air spilled through turbulent-flow cylinder baffles into the low-pressure chamber. Oil cooling air was taken in through the left side of the nose cowl entrance scoop and ducted back to the oil cooler. The carburetor air entered at the bottom of the nose cowl and was ducted between the rows of exhaust stacks and made one 90-deg bend up to the carburetor.

The maximum cylinder-head and barrel temperatures encountered in test are shown in Fig. 22. Cylinder No. 6, left bank, lee side, was consistently hottest. Base temperatures were marginal. Of interest, however, is the fact that the base temperature spread is a mere 7 F.

In the C-1B installation, controllable cowl flaps opened to 4-in. maximum gap at the trailing edge of flap segments, a flap angle of approximately 35 deg, and closed in the faired position to 1 in. minimum.

XP-77 – The XP-77 has a common entrance and pressure box for engine cooling air, oil cooling air, and carburetor air. Separate ducts at the rear of the pressure chamber bleed off the carburetor and oil cooling air.

Full-scale wind tunnel investigations were conducted by the NACA to determine the cowling and cooling requirements of the SGV-770 engine as installed in the XP-77. Two cowling configurations, one with a triangular entrance *A* (Fig. 23A), and the other with a circular entrance *B* (Fig. 23B) were investigated. The plane of the



■ Fig. 8 – Static pressure distribution over side of nacelle – low wing

triangular entrance was sloped downward and aft 2 deg from the plane of the propeller; the plane of the circular entrance was sloped downward and aft 14 deg.

An unusually high inlet ram of 1.45q for the simulated climb condition was obtained with the triangular entrance. This was attributed to the piling up of air ahead of the cowl entrance, resulting from the blocking effect of the fuselage and the effect of the propeller in front of the entrance.

The effects of inlet velocity ratio and angle of attack on the front cylinder pressures are shown in Figs. 24 and 25. The presence of the oil cooler in the installation with the circular entrance, as shown in Fig. 23B, restricted the airflow, causing flow velocities in the V of the engine which resulted in low-pressure recoveries.

Provision was made for retracting the nose wheel gear into the V of the engine. Although it was believed that the obstruction might seriously impair the cooling of cylinder Nos. 1, 2, and 3 of both banks, ground cooling tests showed that satisfactory cooling could be achieved.

The engine cooling-air exits for the two cowling configurations *A* and *B* are shown in Figs. 26A and 26B, respectively. In *A* the exits were located opposite a point midway between cylinder Nos. 3 and 4; these exits incorporated two vanes each to direct the exit airflow parallel to the flight path. In this installation a sliding door replaced the conventional cowl flap at the cooling air exit.

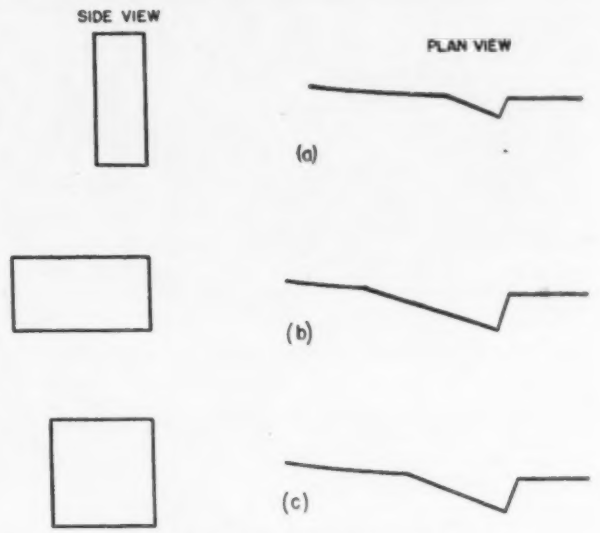
The *B* installation exits were located well aft near the firewall. For the simulated climb conditions with the *A* installation, the pressure at the cylinder baffle exit was 0.31q_0 higher than the cowl exit static pressure; with the *B* installation under similar conditions this difference ran from 0.60q_0 to 0.77q_0.

Inasmuch as both exit areas were substantially the same with flaps open, it is believed that the lower pressure drop of the *A* installation was caused by the central location of the flaps, and the fact that the *B* installation exit was influenced by the flow restrictions upstream of the exit.

Work on the XP-77 was concluded after completion of the performance flight tests. However, in order to determine that safe cooling would be achieved during the performance flight tests, a few cooling tests were conducted. These tests revealed that all cylinders cooled considerably under the maximum allowable temperatures for both cylinder heads and bases.

XAT-6E – The XAT-6E airplane represents a modern in-line installation and incorporates design characteristics based on wind tunnel investigations and flight tests with previous V-770 powered aircraft. As shown in Fig. 27, the high-pressure chamber is a common reservoir for engine cooling, oil cooling, and carburetor air, supplied through a common entrance. It extends across the entire lower portion of the cowl and back to the rear vertical baffle. The rear vertical baffle, located just aft of the rear cylinder, extends across the cowl and thus separates the accessory compartment from the high-pressure chamber. A horizontal baffle extending fore and aft and from the engine crankcase to the cowling completes the enclosure of the low-pressure chamber, and completely isolates the accessory compartment from the cylinder cooling air. Oil cooling and carburetor air are taken through entrances in the rear vertical baffle of the high-pressure chamber.

Design considerations resulted in an engine cooling-air exit flap, shown in Fig. 28, with a span to chord ratio of nearly one, a ratio higher than considered optimum. This large exit area was established as a requirement for satisfactory cooling at high altitudes; subsequent study indicates probable reduction of the exit area and the flap size.



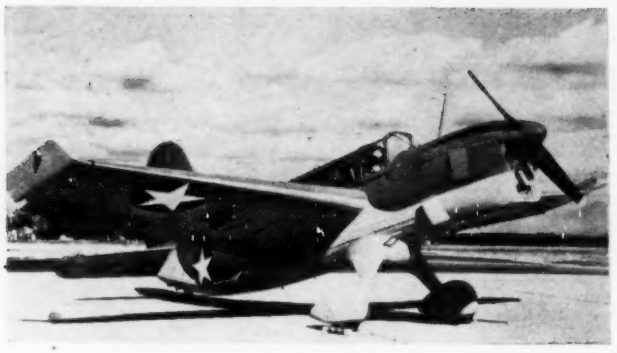
■ Fig. 9 – Exit shapes tested in wind tunnel

Average total pressure recoveries in climb varied from 1.55q at 10,000 ft to 1.3q at 25,000 ft. The total pressure was measured with a shielded total head tube at the rear of the V. The cowl flaps worked exceptionally well, a three-fourth open position of the flap proved sufficient for cooling the engine in climb. The flaps induced, at the lee side of the baffles, a static pressure ½q below free stream static pressure. This resulted in a pressure drop across the engine of 10 in. of water in climb up to the second critical altitude.

Internal drag was effectively reduced; the total internal ducting losses amounted to only about 10% of the engine pressure drop. Comparison of flight test results obtained on this installation with those obtained on the original radial installation indicates that a marked reduction in overall powerplant drag is realized with the in-line installation.

The cylinder-head and barrel temperatures encountered in normal rated power level flight at 10,000 ft in auto-lean operation are presented in Fig. 28. Ample margin is apparent between existing and maximum allowable temperature limits. Auto-rich rated power climb tests with this installation displayed marked over-cooling with the cowl flaps only three-fourths open at 14,000 ft. This was attributed to use of a flap opening far exceeding optimum for the particular flight condition.

General Comments – In addition to the development of the Al-Fin cylinder and the turbulent-flow baffles a third measure, concerned with the methods of delivering the cooling air to the cylinders, is now under investigation. This system is known as "bipolar cooling" and delivers cooling air to both the downstream and upstream sides of the cylinder. Air enters the baffled zone (Fig. 29) from both the inboard or 0-deg position and the outboard or 180-deg position. Air from both sources is exhausted at the 135-deg position which is aft, 45 deg outboard. It is expected that installations employing bipolar cooling will



■ Fig. 10 – Curtiss SO3C airplane

■ Fig. 11 – Fairchild AT-21 airplane

■ Fig. 12 – Bell XP-77 airplane

■ Fig. 13 – North American XAT-6E airplane

require approximately the same total cooling airflow as the conventional full-pressure cooling installation. However, because only approximately 80% of the total cooling air must pass completely across the cylinder, the cooling power expenditure is considerably reduced. Jet augmented cooling is likewise currently under investigation in an effort to realize fully any of its advantages.

Also under consideration are the problems of completely submerged engines to meet requirements of a helicopter, flying wing, or in-the-wing installation. A V-770 engine has been successfully installed, completely submerged, in a 36-ft military boat. Cooling air is supplied by an axial-flow fan driven from the propeller shaft.

■ Cowling Arrangements

The cowling may be subdivided immediately into two categories, namely, internal and external. For discussion purposes the external cowling may be further subdivided into six major components: the propeller spinner, the nose cowl, the top section, the bottom section, the side sections, and the cowl flaps. Depending upon the nature of the specific installation, the four cowl sections, particularly the two side sections, are frequently further subdivided for structural reasons.

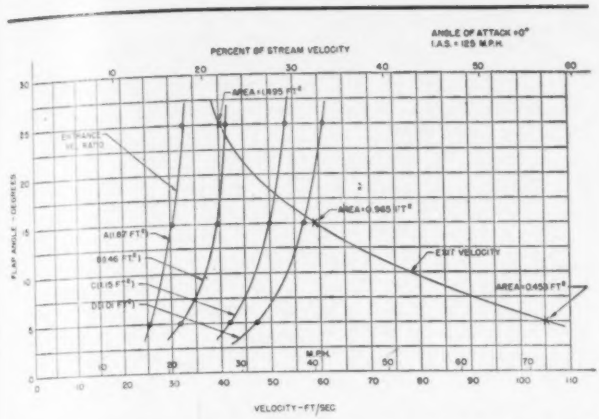
The subject of the propeller spinner is of questionable importance in the case of low-speed and moderately powered aircraft. With the 12-cyl series engines, however, the propeller spinner assumes importance. The high propeller shaft location necessitates use of the spinner to fair in the top line of the cowling, to shroud the larger hub and pitch changing equipment, and to minimize disturbances in the airflow immediately before the main air intake of the aircraft.

Nose cowl design for this type of installation embodies several considerations which increase in importance as engine powers and flight speeds increase. Careful thought is required to establish the most favorable location, size, shape, and fairing obtainable, with the established cowl lines, for the engine cooling air inlet. Inlet entrance lips patterned after suitable NACA airfoils are recommended for use wherever practicable. The nose cowl must fair forward into the propeller spinner and aft into the main cowl. This must be accomplished in such a manner that flow separation and critical velocities will be avoided.

Nose cowl construction has been accomplished in a variety of ways; the details may vary in accordance with the method of support. One of the simpler designs embodies a one-piece formed metal shell or body to which stiffening bulkheads are added. Designs using propeller spinners incorporate a circular pressed-metal bulkhead at the spinner circle which attaches to the nose section of the engine. A second bulkhead at the aft end of the cowl is attached to either the engine or the mount.

Construction of the main cowl usually follows one of two typical designs involving a difference in the treatment of cowl supporting and attaching structure.

One type employs a cowl former structure, usually of channel-section members, suspended from the engine mount and nose cowl. Removable cowl panels are attached to this framework. Cooling-air exit flaps are suspended in the framework, and the removable panels are made small enough to promote ease of handling. Auxiliary access doors are generally provided where frequent admission is necessary. The top section, not as frequently removed, is usually made in two individual panels. The bottom sections present the problem of cowling around the exhaust stacks. A fixed member is installed fore and aft along either row of exhaust stacks; this fairs into the lower cowl. This section is made a portion of the structural framework, and smaller, lower panels are attached to it to complete the lower cowl. A large access door is sometimes provided in the lower cowl to facilitate spark-plug accessibility.



■ Fig. 14 – Entrance and exit velocity ratios obtained from wind tunnel investigations of various cowl entrance and exit areas

The second type of approach to the problem assembles the forming and supporting structure to the cowl skin. This method results in four removable panels which assemble by means of mechanical latches to fittings which are integral with the engine mount. A hinged side panel suspension, illustrated in Fig. 30, is designed to afford speedy access to the engine; the attaching fittings, of the barrel-bolt type, are arranged in line, thus allowing the entire side panel with its integral structure to swing outward to a horizontal position. With this type of structure, cowl flaps are incorporated in the side panel doors and therefore swing with the doors. Cowl flap controls are so arranged that the doors may be opened without breaking the control connections. Top panel construction is one piece and attaches by mechanical latches similar to those retaining the doors.

The design of internal baffles is dictated largely by the specific requirements of individual installations. The pressure box, which is a zone supplied with cold ram air entering the cylinder V through the nose cowl, must be sealed between the cam boxes and the side cowl and aft at the bulkhead, sealing the rear of the V down to the lower cowl. Because of the airflow from the center of the V outboard through the cylinders, two identical low-pressure chambers are required, one on the outboard side of either bank. These zones are scavenged by the pumping action of the cowl flaps.

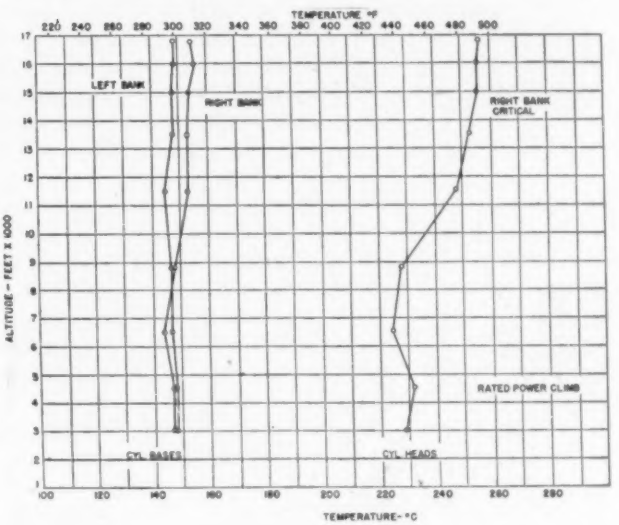
A proposed installation, incorporating bipolar cooling, admits all required cooling and carburetor air through a common inlet in the nose cowl. This entrance supplies two pressure zones, one within the engine V and a second pressure box located above the crankcase. This is supplied by ducts running upward from within the nose cowl entrance. Bipolar cooling air ducts run downward from the overhead pressure zone and blast the outboard side of individual cylinders with cold, high-pressure cooling air. The total cooling airflow to a given cylinder is then discharged into the low-pressure chamber from a point in the baffle, as shown in Fig. 29, 45 deg aft and outboard with respect to the cylinder axis.

■ Oil Cooling Systems

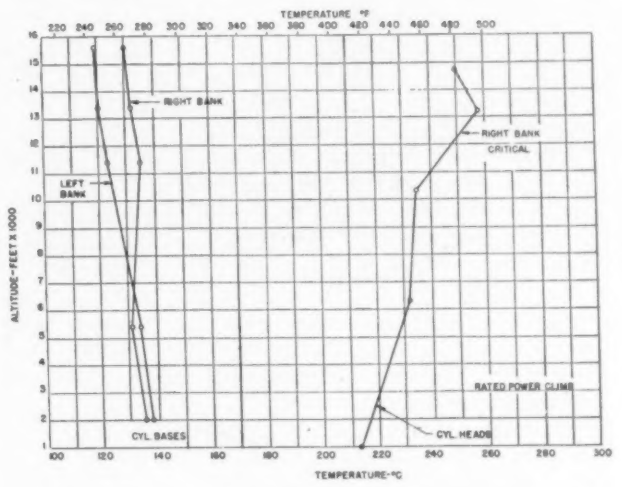
Lubrication system installation considerations singular to in-line, aircooled engine applications are normally confined to the location of the oil tank, oil cooler or coolers, and the handling of the cooling air.

Where space limitations allow, the oil tank is usually located behind and above the rear section of the engine. This location provides the most efficient mounting facility, normally the engine mount members. However, in installations where the engine is necessarily close coupled to the firewall, it is possible to locate the tank above the crankcase of the engine. When this is done, some penalty in tank weight is sacrificed because its shape must be such as to fit saddlewise over the engine and hence does not permit minimum weight design. However, it does permit mounting the oil tank in the engine compartment, which is desirable in an integral powerplant unit.

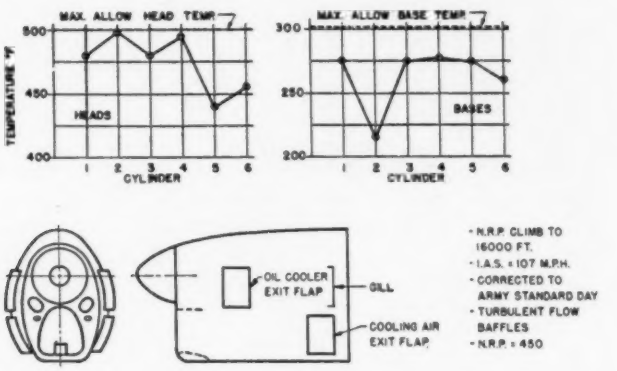
Oil cooler location in existing installations of the 6-cyl series engines has frequently been a matter of convenience. Generally, the oil cooler, which is relatively small, is located in the readily available space above the engine cooling air pressure box. Secondly, the proximity of the nose cowl renders oil cooling air supply a relatively simple problem. Disposal of the warm air exhaust from the



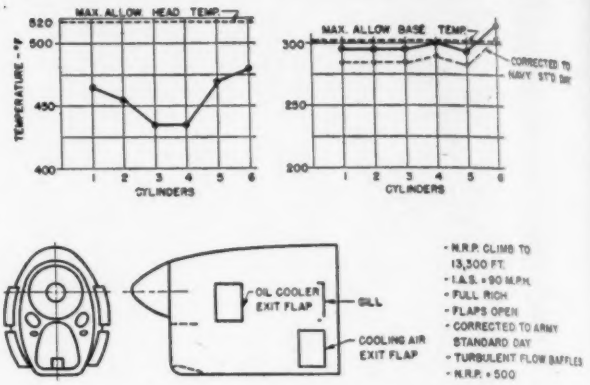
■ Fig. 15 – Temperature variation with altitude – engine equipped with standard baffles



■ Fig. 16 – Temperature variation with altitude – engine equipped with turbulent-flow baffles



■ Fig. 17 – Maximum corrected cylinder temperatures (SO3C with SGV-770C-1 engine)



■ Fig. 18 – Maximum corrected cylinder temperatures (SO3C with SGV-770C-2 engine)

cooler is accomplished by a fixed gill of either the clam shell or the louvre type made integral with the side panel.

Subsequent studies indicate the feasibility of an oil cooler installation similar to that used in the XAT-6E described below, which requires little or no ducting. Such an arrangement is considered superior to that discussed above.

On installations of the 12-cyl engines, two general approaches to the oil cooler problem are used. One, usually the superior approach, uses only one cooler so located as to minimize duct lengths. An example of this installation is illustrated in the North American XAT-6E airplane. Here, one of the inherent advantages of the inverted V, in-line engine is utilized. The oil cooler is suspended beneath the engine rear section in the accessory compartment, as illustrated in Fig. 31. With the air inlet face of the cooler located in the plane of the rear bulkhead of the pressure chamber, cooling air is drawn from the engine cooling air pressure box. From the exhaust face of the cooler a duct leads the air overboard through a manually controlled exit flap located in the engine compartment at the trailing edge of the accessory cowl. Table 2 presents flight test performance data on this oil cooling system.

Table 2 – XAT-6E Flight Test
(Total pressures taken at front face of oil cooler)

	Oil Cooler T. H. Front	Ias, mph
Normal Rated Power Level Flight at 14,000 Ft	16.5	170
Normal Rated Power Climb at 10,000 Ft	10	115

The second typical oil cooler installation utilizes two oil coolers operating in parallel with respect to airflow, thus distributing the total required capacity over two units. In installations of this type one cooler is suspended on either side of the engine, either immediately above the cylinder bases just outboard of the crankcase or below the aft region of the camboxes. Twin air openings in the engine nose cowl supply air to the inlet face of the coolers when located alongside the crankcase. The exhaust air from the coolers flows into the low-pressure chamber of the cowling and discharges through exit air flaps. When coolers are located beneath the camboxes, ram air is available from the adjacent pressure box. Oil cooler exhaust air is led overboard through short separate ducts terminating in controlled exit flaps.

The Fairchild AT-21 oil cooler installation represents a combination of two of the methods previously described. This airplane takes in oil cooling air from a point immediately inside the engine cooling air entrance and ducts it aft just outboard of the left-hand cylinder bank. The duct is continuous through the bulkhead located just aft of the rear cylinders back to the air inlet face of the cooler. The airflow through the cooler is directly aft into the exhaust duct, which directs it downward and outboard where it discharges overboard. This installation incorporates an airflow control temperature regulation type of cooler, hence a controllable oil cooling air exit flap is not used.

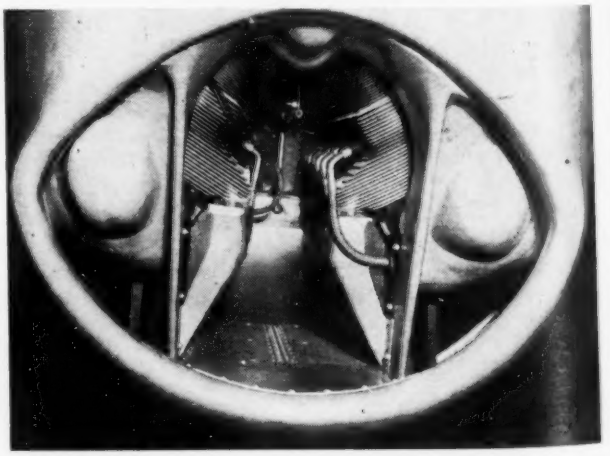
Table 3 presents performance data on the AT-21 oil cooling system.

Table 3 – AT-21 Flight Test
(Total pressures taken at front face of oil cooler)

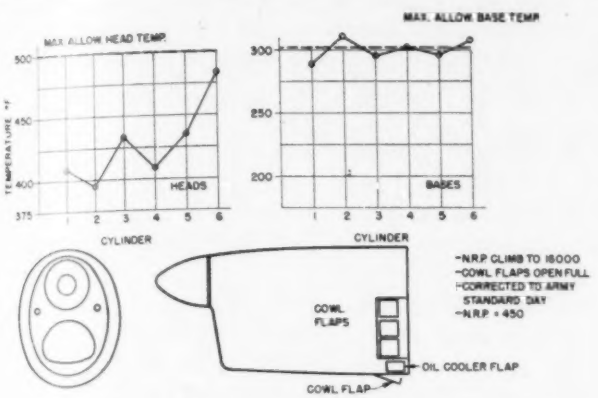
	Oil Cooler T. H. Front	Ias, mph
70% Power Level Flight at 12,000 Ft	11.7	155
Normal Rated Power Climb at 12,500 Ft	6.5	112

■ Induction Systems

That portion of the engine induction system for which the installation engineer is responsible consists essentially of three simple but important components, the air entrance, the connecting air duct, and the charge air elbow. Various types of additional air conditioning equipment become



■ Fig. 19 – Air entrance XAT-14 nacelle



■ Fig. 20 – Maximum corrected cylinder temperatures (XAT-14 with SGV-770-1A engine) (experimental installation – see Fig. 22 for production version)

necessary in specific applications; this equipment will be mentioned later.

Structurally, there arise the ever-present considerations of light weight and resistance to fatigue under vibration. Long induction systems pose the problem of incorporating the strength necessary to withstand effects of backfire.

An analysis of the more specific considerations arising in conjunction with induction system design begins with the air entrance. Entrance areas should be established as the minimum area, in keeping with satisfactory inlet velocity ratio, required to meet both the maximum power level flight at critical altitude and the maximum power climb at critical altitude. A check should also be run covering take-off conditions. The design of the duct entrance shape and the fairing around the entrance are dictated so closely by the location and the particular installation under consideration as to be beyond the scope of this discussion.

That portion of the duct carrying the air from the entrance to the carburetor elbow or charge air elbow assembly in many installations performs the additional function of a diffuser. Diffusers in general seem to behave most satisfactorily when the included angle of divergence is held between 6 and 15 deg. The controlling factor within this range appears to be duct aspect ratio; in general, as the aspect ratio is reduced, the included divergence angle must be reduced.

The importance of the diffuser is somewhat diminished in view of recent efforts to accomplish as much as possible of the conversion from kinetic to pressure energy in the air stream prior to induction of the air into the duct. This generates the concept of constant velocity ducting which will minimize the losses within the duct because of the absence of conversion losses.

Auxiliary equipment, primarily for air conditioning purposes, is installed in accordance with aircraft specifications arising from the type of service intended. Addition of this equipment tends to modify design of the several induction systems' major components. Most commonly and radically affected of these is the carburetor elbow which generally incorporates the carburetor air preheater valve providing for selection of either hot or cold induction air as required.

Carburetor air cleaners of two principal types, the mechanical filter and the viscous impingement type, are installed at points upstream of the carburetor to eliminate solid particles of foreign material from the combustion air.

An induction system displaying favorable performance in flight is that of the North American XAT-6E incorporating a Ranger SGV-770D-4 engine. This system (Fig. 31) draws carburetor air from the aft portion of the pressure chamber through ducting opening in the pressure bulkhead. In normal rated power level flight at an altitude of 14,000 ft, this system attained a pressure recovery of 104% free stream dynamic head. In normal rated power climb at an altitude of 10,000 ft a pressure recovery of 130% free stream dynamic head was obtained. Measurements were made at the air entrance to the carburetor or carburetor deck.

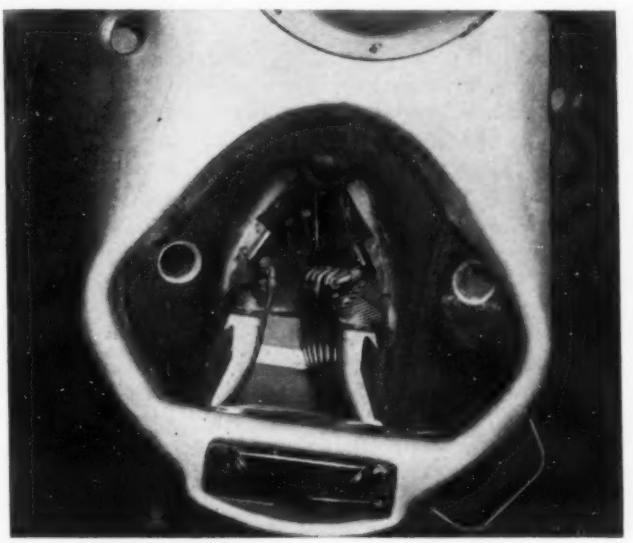
The induction system of the Bell XP-77 (Fig. 32), although it represents a slightly different ducting arrangement, also takes its carburetor air from the pressure chamber within the engine V. It consists of a duct, opening in the pressure chamber at a point about midway between the nose cowl and the pressure bulkhead, which passes aft beneath the cambox of the right bank and turns aft into the carburetor elbow. In high-speed level flight at sea level this system attained a conversion of 114% free stream dynamic head; conversion in rated power climb at sea level amounted to 145% free stream dynamic head.

Conversions as high as 200% free stream dynamic head have been obtained with propeller cuffs for the climb condition at sea level during full-scale tests of a system consisting entirely of a simple carburetor elbow in which the air entered at the rear pressure box bulkhead.

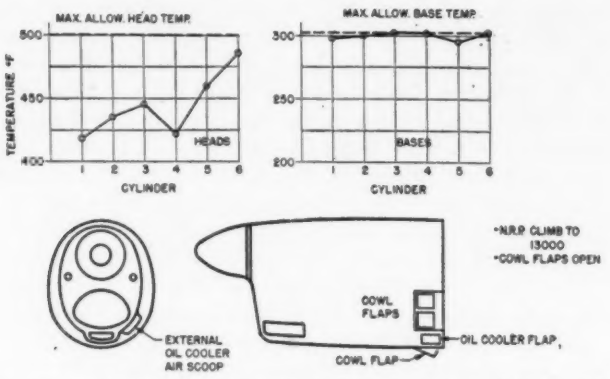
Noteworthy is the excellent pressure conversion in climb realized with these installations. This is attributed to the location of the carburetor air inlet which follows closely the arrangement judged most satisfactory.

■ Engine Mounts

The engine mount transfers the loads resulting from engine operation, installation weight, and inertia loads to the airframe. Generally, four-point mount suspension is employed with points located as close to the cowl skin at the firewall as practical considerations will permit in order



■ Fig. 21 – Air entrance of AT-21 nacelle



■ Fig. 22 – Maximum corrected cylinder temperatures (AT-21 with SGV-770C-1B engine)

to promote maximum strength, suitable rigidity, and convenience in transmitting the loads to the airframe. The fundamental requirements of maximum strength-weight ratio, rigidity, suitable vibration characteristics, and elimination of interference must be observed.

The in-line mount may be considered as a pair of semi-cantilever beams which pick up the four engine crankcase mount pads, one fore and one aft, on either side of the engine, and transfer the loads to the airframe at the firewall.

Stress analysis on mounts of the in-line type is probably not as well developed as for the radial engine mounts. However, suitable methods which have been substantiated by static tests are available. In-line engine mounts are comparable to radial engine mounts on a weight per horsepower basis.

Further progress in weight reduction may result from application of machine-finished magnesium I-sectioned forgings attaching at the upper bolt on either side and supported by compression members tying to the main member adjacent to the aft engine crankcase mounting pad. Construction of this type will be recalled in use on several German liquid-cooled military powerplant installations.

Several mounts carrying the V-770 series engines, all similar in general design aspects, are presently in use. The Fairchild AT-21, the North American XAT-6E, and the Bell XP-77 all use forged mounts. Other installations have been built with welded steel tube mounts.

The Bell XP-77 employs a forged magnesium main beam to which longitudinal stiffeners were welded. The XP-77 airplane is equipped with a tricycle landing gear, and the nose wheel loads are carried through the engine mount which necessitated attaching the nose wheel supporting structure thereto. Incorporating all the above discussed provisions for additional loading and structural attaching fittings, the weight of the XP-77 engine mount proper was held to 45 lb.

A relatively low inertia unbalance, coupled with a high-frequency, low-amplitude power impulse curve, minimizes engine linear vibration in the Ranger 12-cyl series engines. Propeller vibration studies indicate no undesirable exciting forces in the engine affecting propeller performance. Hence, the V-770 series engines have been installed without benefit of any vibration mode decoupling or shock mounts.

The numerous problems attendant with installation of a flexibly mounted, inverted in-line engine are under consideration. Included in the study are questions of cowl construction and attachment, and other support systems. These studies are too preliminary at present to forecast accurately their outcome.

■ Exhaust Systems

The exhaust system of the in-line installation should fulfill the same general requirements established for all aircraft exhaust systems. To review briefly, the primary requirements are listed:

1. The system must direct exhaust fumes away from open passenger, crew, or cargo compartments.

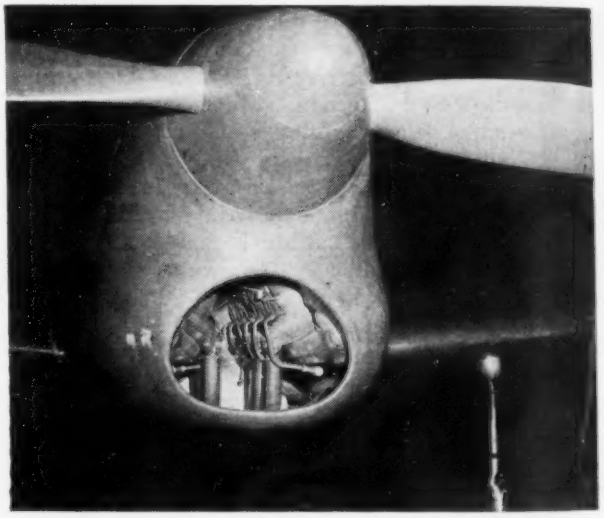
2. The exhaust gases should be conducted and disposed of without danger to surrounding structure.

3. The system should be so located as to minimize fire hazard from overheated or leaking fuel and oil systems.

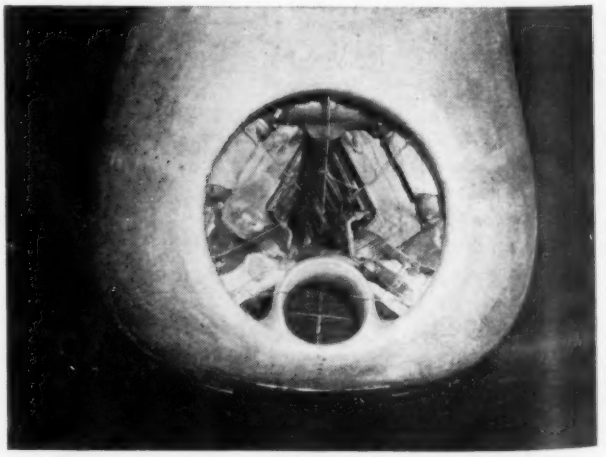
4. The system should not impose undue back-pressure penalties on engine performance.

5. The design should be such that the system will be capable of withstanding normal vibration in service.

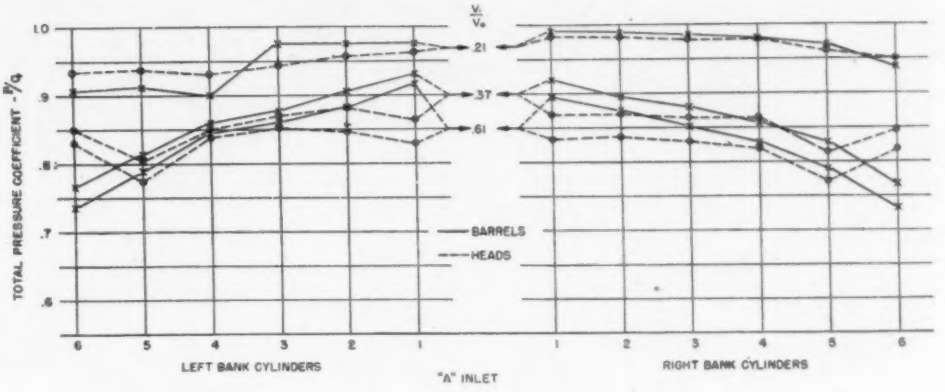
6. The materials should satisfactorily resist deterioration



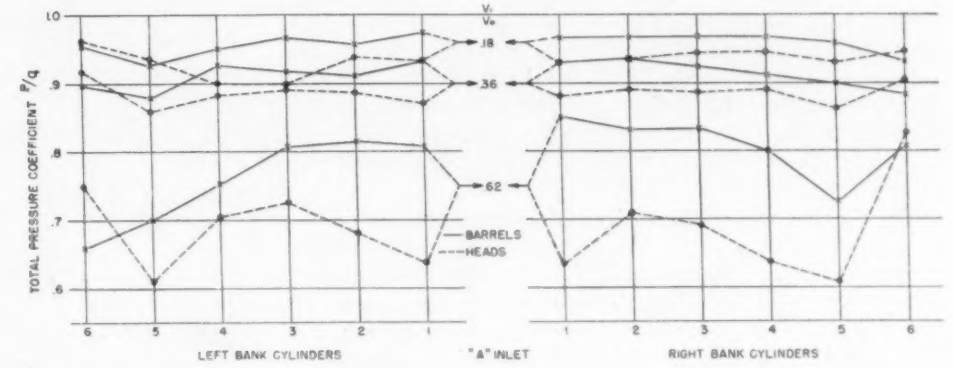
■ Fig. 23A – Air entrance of XP-77 *A* cowl configuration



■ Fig. 23B – Air entrance of XP-77 *B* cowl configuration



■ Fig. 24 – Variation of cylinder front pressures at several inlet-velocity ratios $\alpha = 0$ deg



■ Fig. 25 – Variation of cylinder front pressures at several inlet-velocity ratios $\alpha = 11$ deg

resulting from high temperatures, corrosion, and vibration.

The manifolds used in most installations of both the 6- and the 12-cyl engines are constructed of either welded stainless steel or welded low carbon steel to which an enamel finish is sometimes applied.

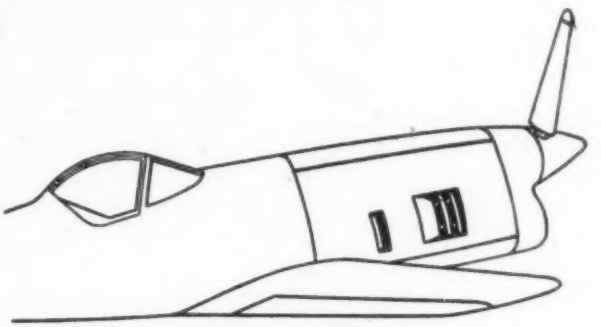
The 6-cyl series engines generally employ a manifold or header with six individual flanged pipes bolting to the cylinder exhaust ports located on one side of the cylinders. The overhang resulting from the side location of the exhaust port flanges necessitates additional bracing to provide adequate support for the header. A single slip joint is generally required between cylinder Nos. 3 and 4 in order to accommodate thermal expansion.

Collectors have been used in 6-cyl installations having two separate assemblies of three cylinders each. Support from the exhaust port pads has been adequate in these cases.

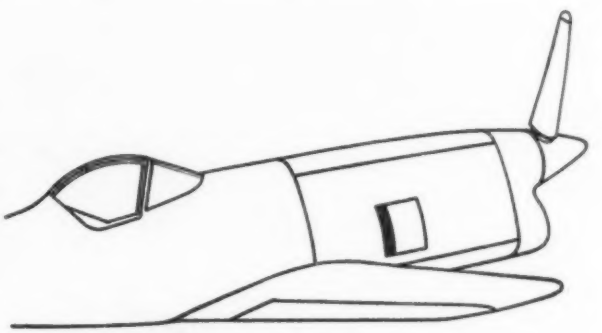
A relatively simple treatment of the exhaust gas disposal problem results from the favorable exhaust port arrangement inherent in the 12-cyl engine.

From the pilot visibility standpoint, with both single and multiengine aircraft, the "belly" exhaust appears satisfactory. With the in-line cylinders inverted, and exhaust ports opening downward from the cylinder heads, a simple, light-weight exhaust system performs the manifold functions of collecting the exhaust gases from the several cylinders, conducting these gases to a suitable location, and disposing of the gases to the air stream.

Considerations of the utilization of exhaust gas jet reac-



■ Fig. 26A – *A* cowl-vane sliding door type of exit



■ Fig. 26B – *B* cowl-flap type of exit

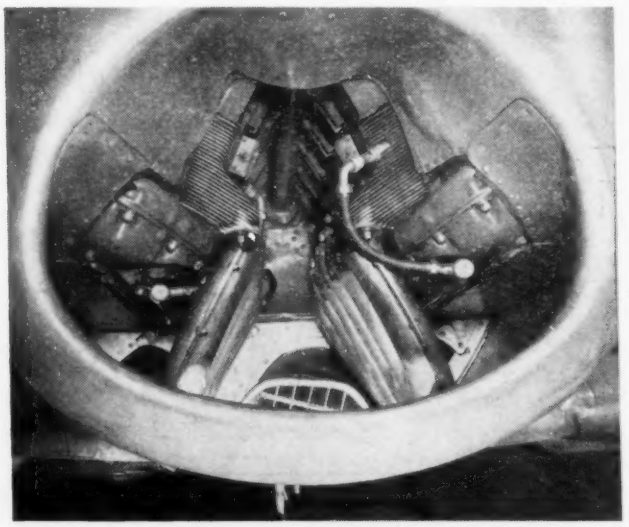
tion and exhaust manifold tuning are frequently displaced by the relatively inflexible space and weight requirements. Exhaust gas discharge should occur parallel to the flight path in order to realize the maximum obtainable benefit.

In the in-line installation, the counterpart of the large "dishpan" and shroud combination comprising the collector well of the more familiar radial installation is replaced by a light sheet aluminum shroud surrounding the exhaust tubing from a point adjacent to the exhaust port outboard to the cowl skin. The space between this shroud and the exhaust tubing, a distance of about ¼ in., allows air, delivered to the shrouds from the pressure box within the V, to flow overboard along the exhaust tubing, thus cooling the stacks themselves and prohibiting interference with engine cooling resulting from the presence of the hot exhaust pipe in the inlet cooling air.

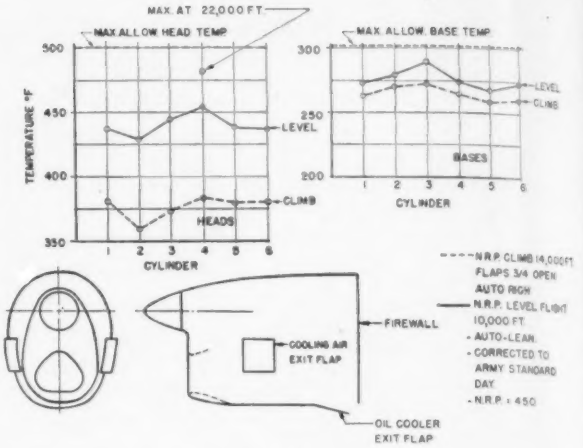
Because of the light weight of this type of exhaust system, the result of both short passages and sheet metal construction, the support problem is likewise simplified. Current installations find suitable support in the cylinder exhaust port attachments. Need for deviation from this arrangement is doubtful unless necessitated by additions to existing systems, such as silencers or flame arrestors, which, from the viewpoint of passenger comfort, are under consideration.

The thermal expansion problem, of considerable importance with collector rings, is usually simplified with the in-line manifold. A recent installation on the inverted V-12 eliminates this problem through use of an exhaust system consisting of four independent manifolds each serving three cylinders. See Fig. 31. Two stainless steel stampings, welded together, with attaching fittings and shroud added, provide a clean, light-weight exhaust disposal system. The weight comparison indicates this system to be only about 66% of the weight of a conventional stub stack installation on a radial engine of comparable power.

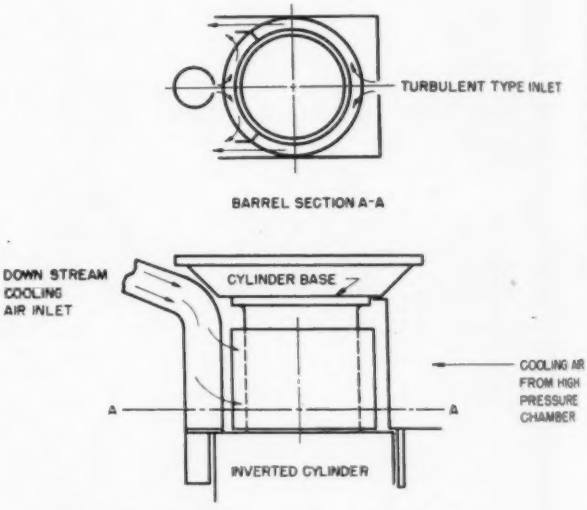
The treatment of installation problems on carburetor air preheaters, cabin heaters, or wing de-icing heaters, may be handled in a manner closely paralleling radial engine practice. The AT-21 airplane is equipped with a carburetor air heater utilizing a larger closed shroud enclosing the stacks from one cylinder bank. Ram air enters the front of this shroud and is ducted from the rear of the shroud to the carburetor mixing chamber. By similar methods, warm air is available from the remaining row of exhaust stacks, for cabin heating and the like.



■ Fig. 27 – Air entrance XAT-6E



■ Fig. 28 – Maximum corrected cylinder temperatures (XAT-6E with SGV-770D-4 engine)



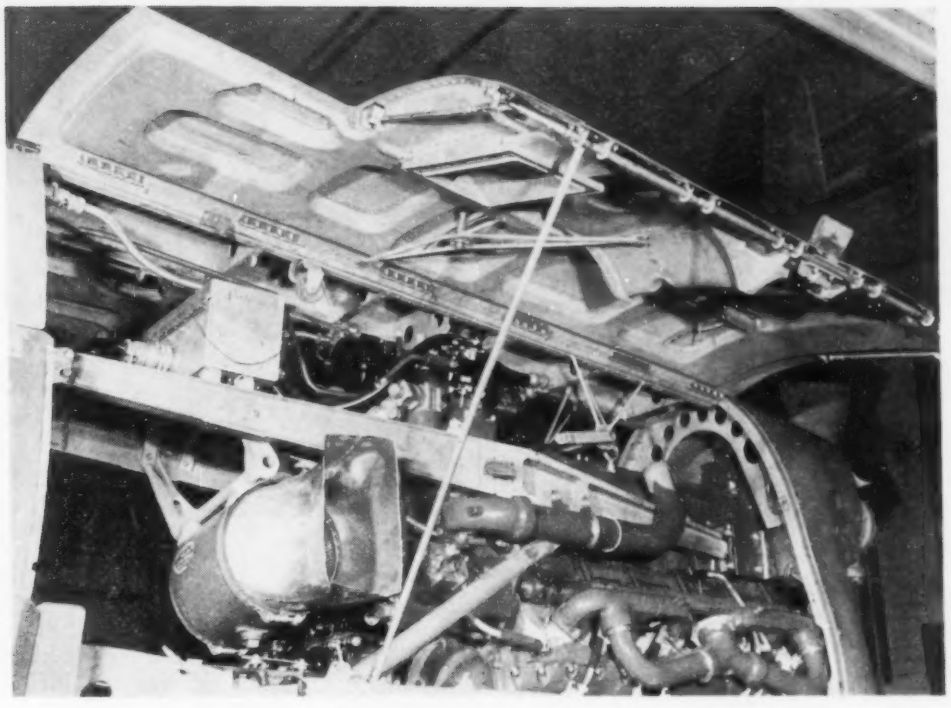
■ Fig. 29 – Bipolar cooling – general principle

The 6-cyl series engines have utilized the commonly used intensifier tube passing through the center of a straight header type manifold to provide hot air for cabin heating. The tube proper opens at the forward end to the air stream, forming a blast tube, and thus provides a warm ram air supply to the cabin.

Any need for exhaust silencers and flame arrestors in the in-line installation introduces the usual problems, namely, increased exhaust back pressure, increased weight, and reduced accessibility.

■ Concluding Remarks

It is evident that sufficient data are now available to design efficiently installations incorporating in-line engines and to predict their performance with reasonable accuracy.

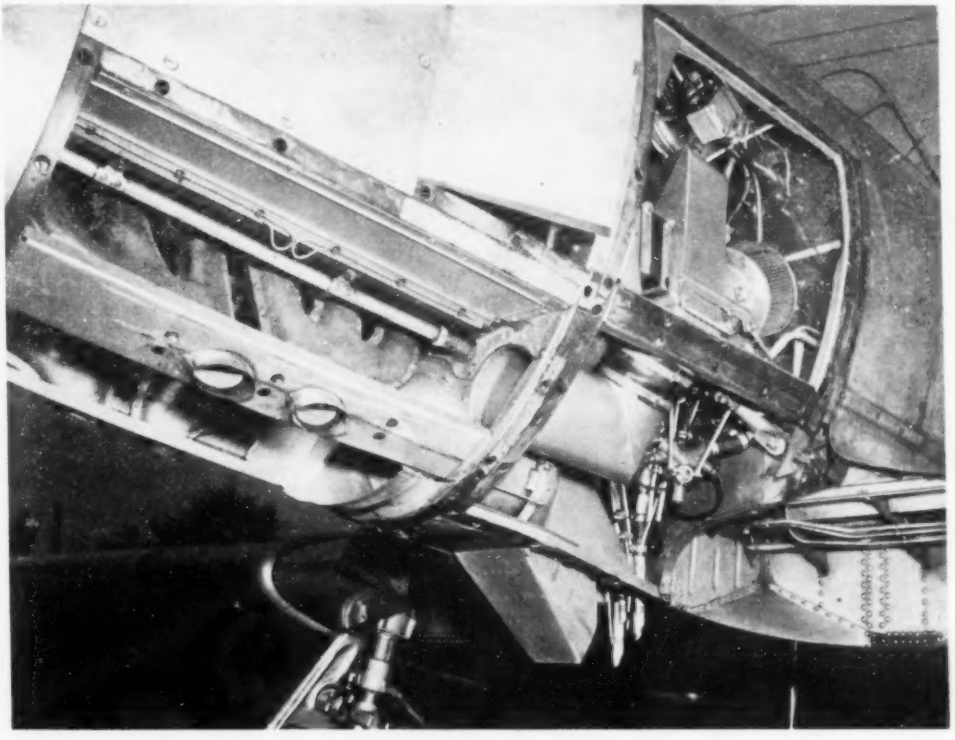


■ Fig. 30 – Hinged side cowl of XP-77

It is also felt that the material presented will be of assistance in connection with such work. The material will be most beneficial from the standpoint of design principles, as it is too early to expect that a single installation is suitable for a given class of aircraft. It is hoped, however, that custom powerplants and components thereof will prove readily adaptable to a wide range of aircraft in the not too distant future.

The present status of in-line, aircooled engine installation design has been brought about by considerable effort on the part of many companies, governmental and other agencies, and their staffs. The most outstanding of them, in addition to the Army, Navy, and NACA, are Bell, Curtiss, Fairchild, and Grumman aircraft companies, and the Massachusetts Institute of Technology.

Considerable assistance in the preparation of this paper



■ Fig. 31 – Left-side view of XAT-6E engine installation

was given by P. B. Freeman, E. Hart, M. Jacobs, J. C. Shroads, and T. M. Sullivan of the writers' staff, and their help is gratefully acknowledged.

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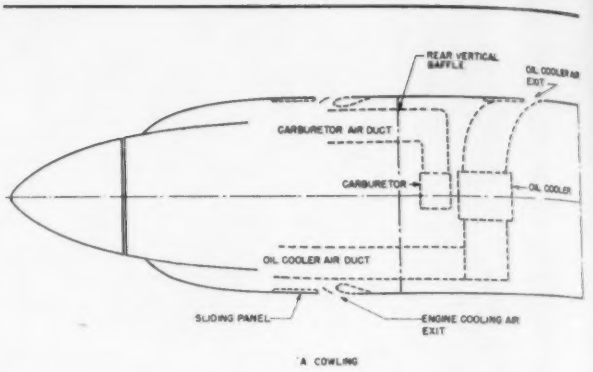

■ Fig. 32 – Diagram of XP-77 induction and oil cooling systems

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Aviation Fuels—What the Airlines WANT and EXPECT

continued from page 132

may be the final answer to greater economy for the air transport industry.

In connection with the above discussion, the following basic thoughts should be kept in mind:

New airplanes will have their payload determined by the volume capacity of the fuselage. Even present-day twin-engined equipment operate under a space-load limitation where short ranges are involved. Introduction of 4-engine aircraft will permit longer range on the same comparative weight-carrying capacity. This is due to higher take-off gross weights being permitted under any given power.

Present landing weight limitations are due to an arbitrary maximum permissible landing speed of 80 mph or to lengthened runways being required as wing loadings increase. The ultimate limitation, however, will always be controlled by the volume capacity of the fuselage.

It has been found to be quite satisfactory and cheap to install powerplants giving us excellent economy at very high cruising speeds. Such installations tend to provide excess take-off powers, which in turn allow a relatively large increase in take-off weight. Thus, high speeds and long-range operations go hand in hand. This is true since the big spread between take-off and landing weights permits excess fuel to be carried. Economy must, in the final analysis, therefore, be based on the cost in dollars per horsepower per hour; provided this low cost is consistent with normal operating practice.

In conclusion, the author trusts that the thoughts expressed in this paper will result in less talk of "super fuels," premium prices, restrictive requirements, et al, and that more down-to-earth, realistic action will be taken by those concerned to develop and give the airlines good quality and reasonably priced aviation fuels.

Effect of DIESEL FUEL ECONOMY on HIGH-SPEED TRANSPORTATION

by F. GLEN SHOEMAKER and H. M. GADEBUSCH

Detroit Diesel Engine Division
General Motors Corp.

IT has been found that the economic formula for automotive transportation is identical with that of other profit-making enterprises, that is, to get the work done at the lowest possible cost.

Transportation work consists of moving the biggest payload over the greatest distance in the shortest possible time. The gross income of this business is expressed, in the case of trucks, as ton-miles per day and in the case of coaches, as passenger-miles per day.

The relative expenses per mile incurred by an average trucking operation using gasoline engines are shown in Fig. 1. The same relative distribution of costs applies equally well to passenger transportation.

As can be readily seen, the daily recurring cost of the fuel represents the third largest item.

For this reason, the ton-miles or passenger-miles operated per gallon of fuel have become an important barometer for the economics of commercial transportation. It is this item of fuel cost that will be discussed as it affects operating costs and vehicle performance with particular reference to diesel engines.

History

Looking back over the last two decades of gasoline-engine development for commercial vehicles, three distinct phases may be discerned.

Early heavy trucks were powered by specially built, low-speed gasoline engines. These heavy and space-consuming powerplants, operating at speeds around 800 to 1200 rpm, served their purpose well but possessed the drawbacks of low production volume, handfitted parts, and consequently, expensive repairs.

As next step, the rapid growth of the automobile industry in the early twenties made light-weight passenger-car engines available as attractive successors.

Much could be said for the use of stock engines. Their installation dimensions were small and their price low. All replacement parts were readily available, inexpensive, and completely interchangeable.

The power output of these gasoline engines was sufficient to move moderate loads at the then prevailing road speeds. Furthermore, their power and speed increased steadily from year to year as result of passenger-car developments.

Fig. 2 illustrates this situation with the example of a

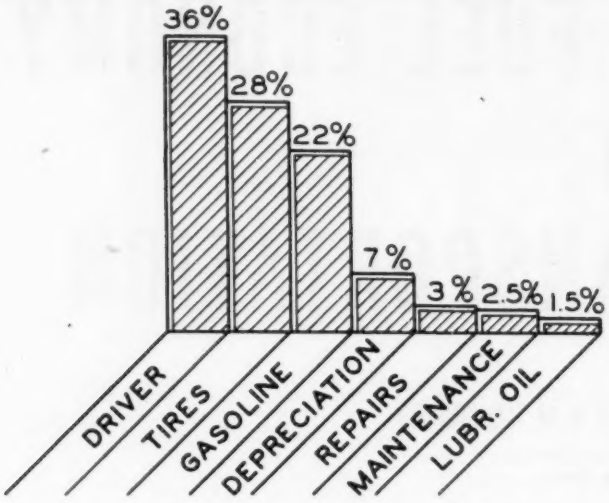
[This paper was presented at a meeting of the St. Louis Section of the SAE, St. Louis, Mo., May 15, 1945.]

USING gasoline engines as a source of power, optimum fuel economy is obtained by operating the engine at a high average load factor, according to the authors of this paper.

Their analysis of the available reserve power of a tractor-trailer combination using a passenger-car engine as powerplant, reveals that the road performance of such under-powered vehicles is insufficient for high-speed transportation. Replacement of the inexpensive small engine, in turn, with a more powerful, special heavy-duty engine will result in an appreciable decrease of the fuel economy on account of the then prevailing light average load factor.

The authors point out that the automotive diesel engine with its basically 25% to 40% lower fuel consumption permits utilization of high average vehicle speeds by providing up to 25% more reserve power for acceleration and hill climbing, at the same time still producing fuel savings of 25% to 30%.

THE AUTHORS: F. GLEN SHOEMAKER (M '20), a General Motors employee for the past 18 years, has devoted much of his time there to the development of 2-cycle diesel engines. Thus, when the Detroit Diesel Engine Division of GMC was formed in 1936, Mr. Shoemaker became its chief engineer, a post he still retains. Participating in World War I as instructor of airplane engines with the U. S. Army School of Military Aeronautics at the University of Illinois, his alma mater, the author pursued his aircraft interest, excepting a brief interval at the Buda Co., at the Air Service Engineering Division at McCook Field. Mr. Shoemaker is a former vice-president of the SAE Diesel Engineering Activity. H. M. GADEBUSCH, who received his technical education in Germany, has been associated with General Motors Corp. since his arrival in this country in 1925, and he is now a member of the engineering staff of the Detroit Diesel Engine Division.



■ Fig. 1 – Operating expenses (trucking)

typical automobile engine widely employed in commercial transportation.

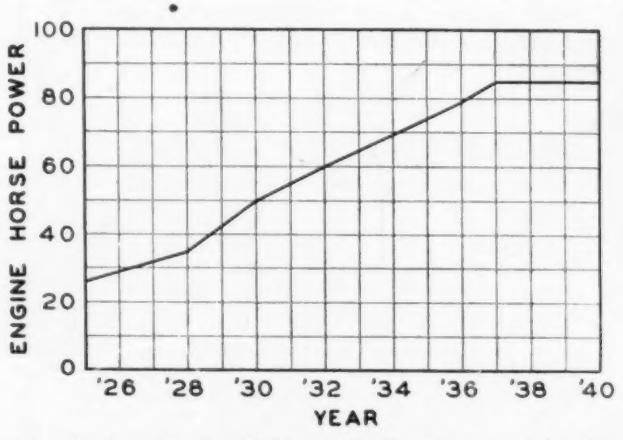
During the years from 1925 to 1937 the brake horsepower shows a continuous rise at the rate of approximately 10 horsepower, for every two years. The curve then levels out, culminating in a total power increase of 227% over the year 1925. Engine speeds in the same period rose from 2000 rpm in 1925 to 3400 rpm in 1940.

Faced with this spectacular provision of more engine performance, commercial operators quickly reacted by boosting the load on their vehicles.

Trucks originally rated by the manufacturers at 1¼ tons were made to carry two, three, and finally four tons of payload until state restrictions on the permissible wheel loading put a stop to further increases.

The operators' desire to take the fullest possible advantage of the available engine power ultimately produced commercial vehicles of a type which has become only too familiar, that is, tractor-trailer combinations powered by a modified passenger-car engine and carrying from 8-10 tons payload.

Fig. 3 illustrates the relations between the road resistance and the available driving power of such a vehicle.



■ Fig. 2 – Increase of available power of automotive engines for the period 1925-1940

The engine load factor, that is, the ratio of average power required to maximum power available, indicates 20% to 50% reserve power for acceleration for theoretical level road driving. However, grades as low as 1% alter the picture completely, necessitating not only a speed reduction to 28 mph but also increasing the load factor to 100% for the remaining part of the power curve.

In actual road operation, the performance of such a combination on average highways involves wide-open throttle operation most of the time.

To cover up the deficiency in power, the manufacturers had to provide more gear ratios, under or overdrives, and dual-ratio axles. Four forward gears and an overdrive became the minimum combination by which the engine torque could be satisfactorily manipulated to meet the varying demands of the highways. These are the trucks which have to shift back to second gear whenever they encounter a 2% or 3% grade, slowing down to 15 or 20 mph, and interfering with all normal traffic movement.

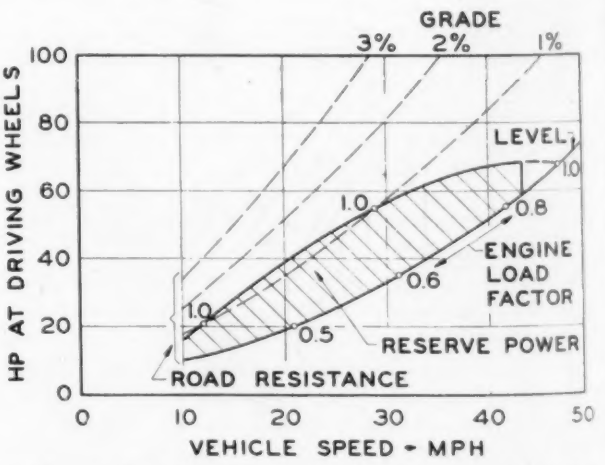
The fuel consumption figures obtained from these low-powered commercial vehicles were good, for it so happens that gasoline engines reach their best economy at or near the full-load point.

While the ton-mile per gallon figures obtained were satisfactory as result of the good engine economy and the high payloads, operators of this equipment began to realize that lack of reserve power prevented them from utilizing the high road speeds of which their vehicles were capable. Furthermore, the continued demand for full power output at high engine speeds created conditions for which passenger-car engines were not designed, and frequent necessity for repairs, excessive wear rates, and low availability threatened to consume a good share of the profits.

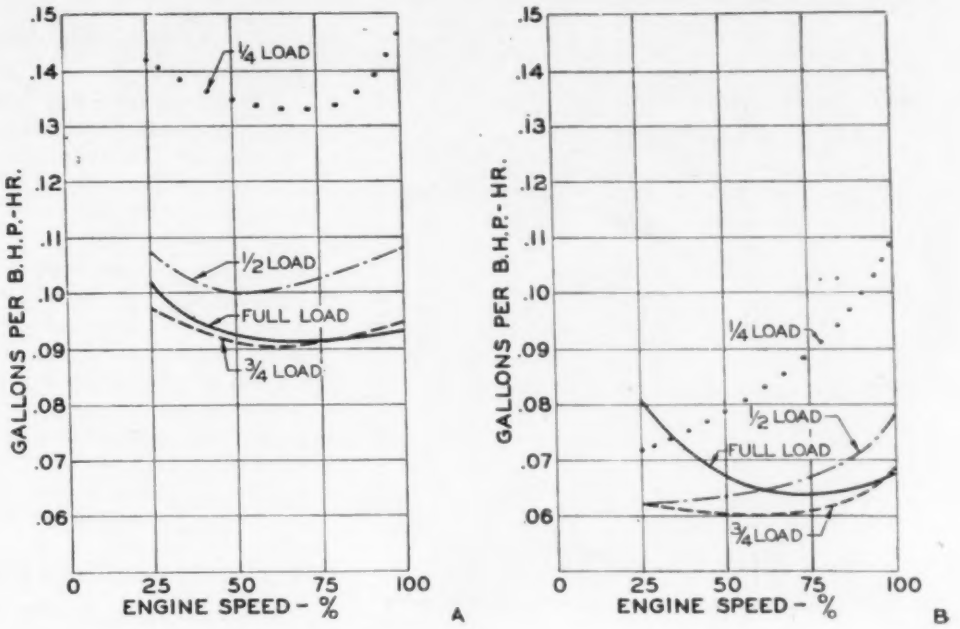
Insistent demands for better engine durability and more reserve power initiated the third step in commercial engine development.

Since passenger-car engines could neither supply the extra power nor the desired long life, special heavy-duty gasoline engines were designed. All critical parts were strengthened, cylinder liners and valve inserts provided, crankshafts hardened, and alloy-type bearings introduced.

With regards to engine life and availability, these heavy-



■ Fig. 3 – Relationship between road resistance and available driving power of tractor-trailer combination powered by modified passenger-car engine



■ Fig. 4 – Specific fuel consumption for 200-bhp engines: *A:* gasoline engine using gasoline of specific gravity 0.72; *B:* diesel engine using fuel oil of specific gravity 0.85

duty gasoline engines made an admirable showing. Overhaul periods of 100,000 miles began to be reached and time schedules could be speeded up where larger engines were used.

The operators were willing to pay the higher price for these specially built engines and liked their high reliability, but in taking advantage of the additional power, the average engine load factor was decreased with a resultant increase in fuel consumption.

For economical operation of these engines, a high load factor was, therefore, desirable. In order to improve the part-load fuel consumption, the engine manufacturers have added special mixture and spark advance controls and improved the overall combustion characteristics to a point where the modern heavy-duty coach or truck engine can be made to give a fuel performance as shown in Fig. 4A. Here the specific fuel consumption is practically uniform from half speed to full speed and only 10% greater at half load than at three-quarters and full load.

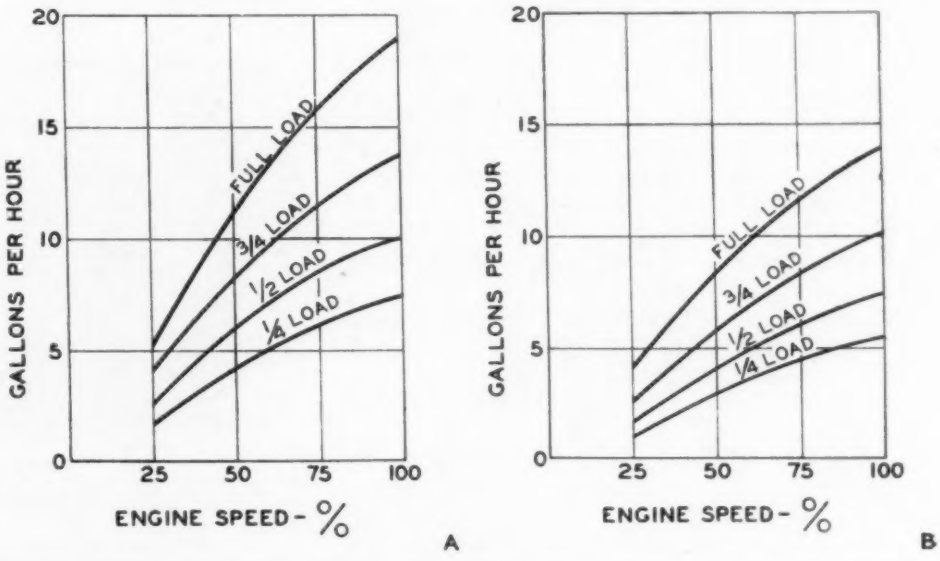
This represents the best gasoline-engine performance commercially available today.

■ Diesel-Engine Economy

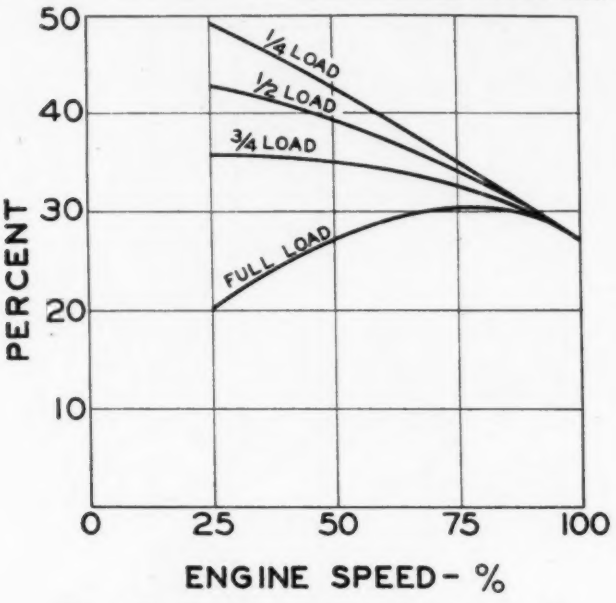
With the advent of diesel engines in the early thirties, the provision of power for commercial vehicles entered a new and fourth phase.

Before being able to compete with the highly developed, light-weight gasoline powerplants, the diesel engine had to overcome many handicaps of its predecessors, such as low speeds, heavy weight, hard starting, noisy running, smoky exhaust, and objectionable odors.

In compensation, the diesel-engine manufacturers were



■ Fig. 5 – Fuel consumption of 200-bhp engines: *A:* gasoline engine using gasoline of specific gravity 0.72; *B:* diesel engine using fuel oil of specific gravity 0.85



■ Fig. 6 – Diesel fuel saving compared with gasoline saving

in a position to offer three important commercial advantages: mechanical simplicity, ruggedness, and greater fuel economy.

Quite naturally, these advantages appealed, first, to the operators of heavy equipment where fuel mileages were low, loads high, and where engine reliability and life were foremost considerations. Thus, in the last decade preceding our entrance into the war, more than 10,000 commercial vehicles, and most of them in the heavy-duty class, were powered with diesel engines.

The higher price of diesel engines was offset by the lower fuel expenses which made up for this extra first cost in a relatively short time.

During this period of intensive development, the mobile diesel engine has undergone a complete transformation from the cumbersome, slow-speed powerplant of stationary fame to a truly light-weight engine. It is interesting to observe the means by which the know-how of the automobile industry succeeded in overcoming the original diesel handicaps.

Operating speeds were increased and have by now well exceeded the 2000 rpm mark. The resulting increase in power together with the application of the two-cycle principle and, lately, use of forced air induction on four-cycle engines have combined to reduce the weight per horsepower to compare very favorably with similar heavy-duty gasoline engines.

The fuel consumption of an up-to-date automotive diesel engine is shown in Fig. 4B.

The position and slope of the curves shows the difference in fuel economy of the two engines. This difference is shown in another way in Figs. 5A and 5B, where the gallons per hour are plotted at various loads over the speed range from 25% to 100% speed.

In Fig. 6 the fuel consumption of the diesel engine has been plotted as a percentage saving of the fuel consumption of the gasoline engine at corresponding speeds and loads. This shows a reduction in fuel consumed of from 25 to 40% over the range from full load and full speed to half load and half speed. This is the usual operating range of commercial vehicle engines. The important point to note is that the fuel saving with the diesel engine becomes greater as the per cent load and speed are decreased.

The diesel engine thus possesses the best fuel consumption characteristics where large reserve power is an essential requirement. To permit rapid acceleration and good hill-climbing ability, trucks and coaches can now be provided with more power and, at the same time, actually improve the fuel consumption.

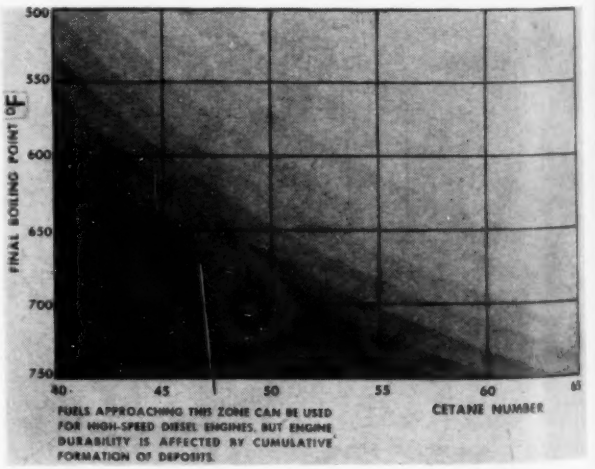
In harnessing the high combustion pressures and temperatures, many of the lessons learned during the development of heavy-duty gasoline engines have been applied. Better understanding of the starting requirements has evolved practical means to supply additional heat to the air charge at temperatures where the heat of compression alone is not sufficient to ignite the fuel. Combustion developments and provision of better fuels have greatly improved exhaust smoke, odor, and cold starting, and lengthened the normal period between engine overhauls.

■ High-Speed Diesel Fuel Requirements

When discussing diesel-engine economics, the fuel price is usually made a major issue. Unfortunately, the name of diesel is still associated in the minds of many people with the use of low-grade fuels and it is not realized that the refinements embodied in high-speed diesel engines have produced a powerplant radically different from its forefathers. The old saying that "diesel engines will run on anything that flows in a pipe" is a deceptive half truth. Operating on the compression-ignition principle, it is true that the automotive diesel engines will burn any combustible material that can be mechanically injected into the cylinder, but the engine still readily distinguishes good fuels from poor ones.

For commercially satisfactory engine life and performance, fuel requirements are similar to those of high-speed gasoline engines, demanding compliance with four basic points:

1. Physical cleanliness.
2. Absence of chemical contamination.
3. Proper burning characteristics.
4. Cold-starting ability.



■ Fig. 7 – Influence of ignition quality and volatility on engine durability

Physical cleanliness means freedom from water, dirt, and other incombustible ingredients. Since all present-day, high-speed engine fuels are completely distilled, they leave the refinery in clean condition. Transport and subsequent storage account for the addition of most foreign matter found in the fuel.

Of the chemical contaminants, the most objectionable are sulfur and gum which, even in relatively small quantities, are largely responsible for harmful internal engine deposits.

The combustion requirements of high-speed diesel fuels are set by the frequent speed and load changes inherent to automotive service which produce rapid variations in combustion-chamber conditions. Cold weather, with the accompanying low intake air, cooling water, and lubricating oil temperatures, serves to emphasize these difficulties.

For satisfactory operation in cold weather, free flow of the fuel to the injectors must be ensured by specification of a low enough pour point. The cetane number must, furthermore, be selected sufficiently high to ensure reliable starting and avoid misfiring at light engine loads.

Consequently, the burning characteristics of automotive diesel fuels as expressed by their ignitibility, range of vaporizing temperatures, and general composition must be more closely defined than those established for large, slow-speed units.

The interrelation of ignitibility and distillation characteristics of fuel oils with respect to engine combustion are graphically shown in Fig. 7.

All fuels in the range of this chart can be burned in high-speed engines with equal thermal efficiency. The available engine life, however, is radically affected by the combination of cetane number and volatility. Fuels in the dark region at the lower left do not burn clean enough for continued satisfactory use. The border line may be drawn at about 50 cetane number and 670 F final boiling point. Following the slope of this curve, it can be seen that equally satisfactory results may be obtained with fuels of 40 cetane rating and 560 F end point – which would be typical West Coast "stove fuels" – and with high-boiling fuels of approximately 60 cetane number and 730 F final boiling point.

A great many of the mechanical troubles experienced in the use of automotive diesel engines are attributable to the use of unsuitable fuels.

Fig. 8 may serve to illustrate this point. The piston shown has been operated for a period of only 6¼ hr at an engine speed of 600 rpm, no load, and with cold intake air and corresponding water temperatures. The fuel used would have passed all accepted standards for stationary diesel fuels. It was a completely distilled material of 35 SSU viscosity, 700 F final boiling point, and 29 cetane number. The picture plainly illustrates the tar-like residue of partially burned fuel working its way from the piston top down towards the skirt.

Fig. 9 shows the heavy carbon incrustations on the exhaust valves of the same engine. Fuels of this type cannot produce satisfactory operating results in high-speed engines.

The important part which fuels and lubricating oils play in the formation of deposits affecting the operating satisfaction derived from an engine is gradually becoming better understood.

Regardless of whether gasoline or fuel oil is used, the normal life of all moving parts in a high-speed engine is

■ Fig. 8 – Tar-like residue covering top of piston taken from engine operated for 6¼ hr at 600 rpm, no load, and cold intake air

greatly reduced by the accumulation of resinous and carbonaceous deposits.

Piston rings stick in their grooves, valve operation becomes sluggish, bearings and piston skirts lose their clearances. Oil control rings get plugged, abrasive hard carbon deposits on the piston lands cause excessive cylinder wall and piston-ring wear and proper lubrication is endangered by sludge formation.

Excellent progress has been made during the last four or five years towards elimination of deposits originating from lubricating oils.

Selection of suitable natural stocks and subsequent fortification by chemical additives have produced heavy-duty lubricants far superior to straight mineral oils.

The high oxidation resistance and detergency of these oils have proved beneficial by avoiding premature engine failures and permitting longer periods between overhauls.

In spite of the initially higher price per gallon of heavy-duty lubricants, the operator's actual oil expenses have been lowered since the longer useful life of these oils permits material lengthening of the oil change intervals.

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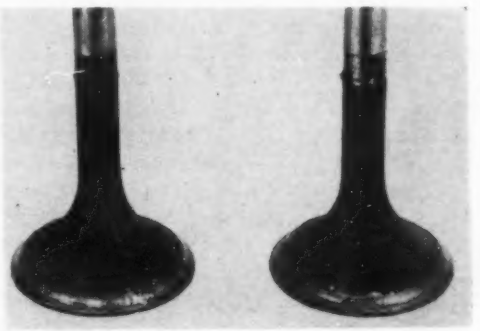


■ Fig. 9 – Heavy carbon incrustations on exhaust valves of same engine as Fig. 8

Application of TURBOSUPERCHARGERS to

THE turbosupercharger, to a measurable extent, owes its development and application directly to the Army Air Forces at Wright Field. Although originally invented by Rateau of France in 1910, the benefits of this device were appreciated by a small group of Army personnel. It was quickly developed in World War I and performed well on both French and American fighters.

Due to continued interest and development by both the Army Air Forces and the General Electric Co., the turbosupercharger was ready to play an important role in World War II. The development in performance, reliability, and production of this device since 1940 has been remarkable. Installed on a military airplane it receives very little attention other than routine periodic inspection. Some of the outstanding aircraft have employed turbos, namely, the B-17, B-24, P-38, P-47, and B-29. In addition to the high-altitude performance of these aircraft, the turbo has improved the fuel economy and range for flight above 15,000 ft. In many instances, especially during emergency operation, powers greatly in excess of anything available with the geared type of supercharging have enabled pilots to escape the enemy or save the aircraft.

With this background, it is only natural to investigate the possibilities of employing turbos in postwar commercial aircraft. For the high-altitude, long-range flight, the turbosupercharger should pay dividends during the postwar era. The turbo design will undoubtedly be modified for this type of operation, and its application will be looked upon in a slightly different light than on military aircraft. The development of the turbine engine, jet engine, and modification thereof will undoubtedly enter the powerplant picture, but the cruising turbosupercharger will fill the transition phase very well.

■ Historical Review

In the presentation of this paper it was thought fitting to review briefly some of the highlights of past Boeing history with respect to turbosupercharged military aircraft. Early in 1933 the idea of a large bomber which would carry a sizeable bomb load over long distances was conceived and presented to the Army Air Forces for consideration. This airplane was known as Boeing Model 299.

The first Model 299 was built in 1934 and employed 830-bhp Hornet engines, and two years later, under contract by the Army, 13 Y1B-17's (Model 299D) were built. These aircraft had 970-hp cyclone engines equipped with the two-speed type of geared superchargers. At the request of the Army Air Forces, specifications were submitted for Model 299E in 1937, which was to incorporate turbosuperchargers. This resulted in Model Y1B-17A in 1938, which was slightly modified and resulted in the prototype of Model B-17B. Undoubtedly, this bomber was to play a high-altitude role in the eventful years to come. A bomber of this type would be relatively free from enemy attack or anti-aircraft fire from the ground. Then followed Model B-17C, D, E, F, and G, each model claiming improvements in both the airplane and the powerplants. Engine power increased from 50-75%, critical altitudes increased to 30,000 ft due to engine, turbosupercharger, propeller, oil system, and fuel system improvements. Basically, the turbosupercharger carried a sea-level engine to altitude and provided the power at relatively unchanged engine fuel consumptions.

While B-17 history was being made, thoughts of even bigger and more formidable bombers were in progress. Basically designed around its forerunner the B-17, which gave such remarkable high-altitude performance, the B-29 was evolved. Turbosupercharging was again the best bet to get this airplane to the high altitudes for speed and bombing performance. Unfortunately, there were no turbosuperchargers of large enough capacity to match the 2200-hp engines to be used in the powerplant. A dual arrangement, using the turbosuperchargers in parallel, was decided upon. At first there was some doubt as to whether stability would exist, since it might be possible for one turbo to

THIS paper presents a brief history of past experience at Boeing with turbosupercharged airplanes. Installation problems are discussed by the author who also offers suggestions for future design.

A basic airplane and engine are chosen and several types of supercharging analyzed by Mr. Disch. A method of turbo selection is also briefly presented by him.

Assuming that flight at 25,000 ft is desired, a comparison of range and payload is shown, depicting the benefits of employing a cruising type of turbosupercharger for increased fuel economy and range.

In conclusion, the author summarizes the overall benefits with this type of installation.

■ ■ ■

THE AUTHOR: F. W. DISCH (M '43) has been employed by Boeing Aircraft Co. since 1941 as powerplant design engineer, most of his duties having been devoted to turbosupercharger installations and performance. A graduate from Stevens Institute of Technology, Mr. Disch has also been affiliated with Case School of Applied Science and the National Screw & Mfg. Co.

[This paper was presented at a meeting of the Metropolitan Section of the SAE, New York City, April 19, 1945.]

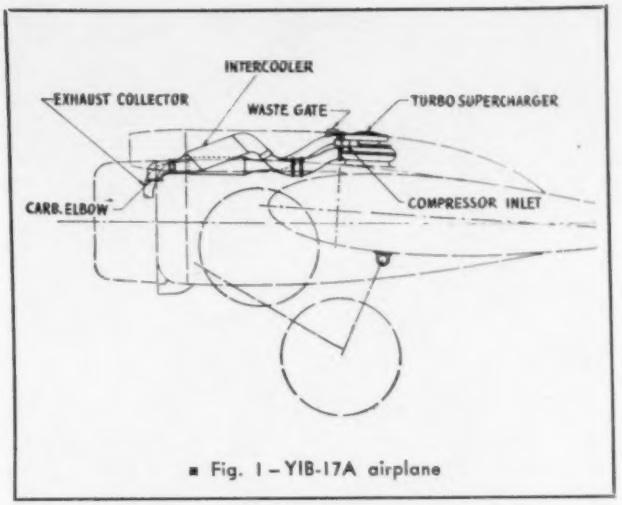
to POSTWAR COMMERCIAL AIRCRAFT

by F. W. DISCH
Boeing Aircraft Co.

overspeed or rob the other of its load. This question was partially answered by the Army Air Forces at Wright Field, which installed a dual turbo installation in a test cell at Wright Field. Although the cell was only capable of creating an 18,000-ft altitude at the turbo exhaust, the installation was inherently stable and satisfactory. However, to test this installation further, a dual turbo installation was built into a B-17 nacelle using reduced capacity turbos. The flight tests showed this system to be inherently stable. The system could be operated either with both turbos or one bypassed by the proper valving arrangements. This feature is beneficial at the lower altitudes and lower cruising powers to avoid a condition of closed waste gate, that is, all of the exhaust gas passing through the turbine nozzles. If altitude is increased at constant power during closed waste gate operation, this must be done by increasing engine rpm, and hence a loss in fuel economy results. However, the power requirements of the B-29 with respect to altitude are somewhat in excess of closed waste gate conditions for the operating schedule used. As the gross weights and hence wing loading are increased, the power requirement will necessarily increase. Thanks to the turbosuperchargers this power is made available at altitude.

On the drafting boards now at Boeing are modified versions of the B-29 powerplant. Larger engines and larger turbos are called for. Past experiences have brought new thoughts into powerplant design. Design engineers have become cooling drag conscious and aware of the importance of many items never even considered in the B-17A days. In preliminary design are bombers more advanced than the B-29. Turbosuperchargers will be considered for some and turbine engines, compound, propeller, and jet types for others. The remarkable characteristics of turbines and the increasing performance with altitude cannot be underestimated. High-speed rotating equipment has an extremely low power-weight ratio, is extremely reliable, and requires relatively little maintenance. The problems of suitable high-temperature metals are gradually being surmounted, as evidenced by the fine performance shown by present turbines.

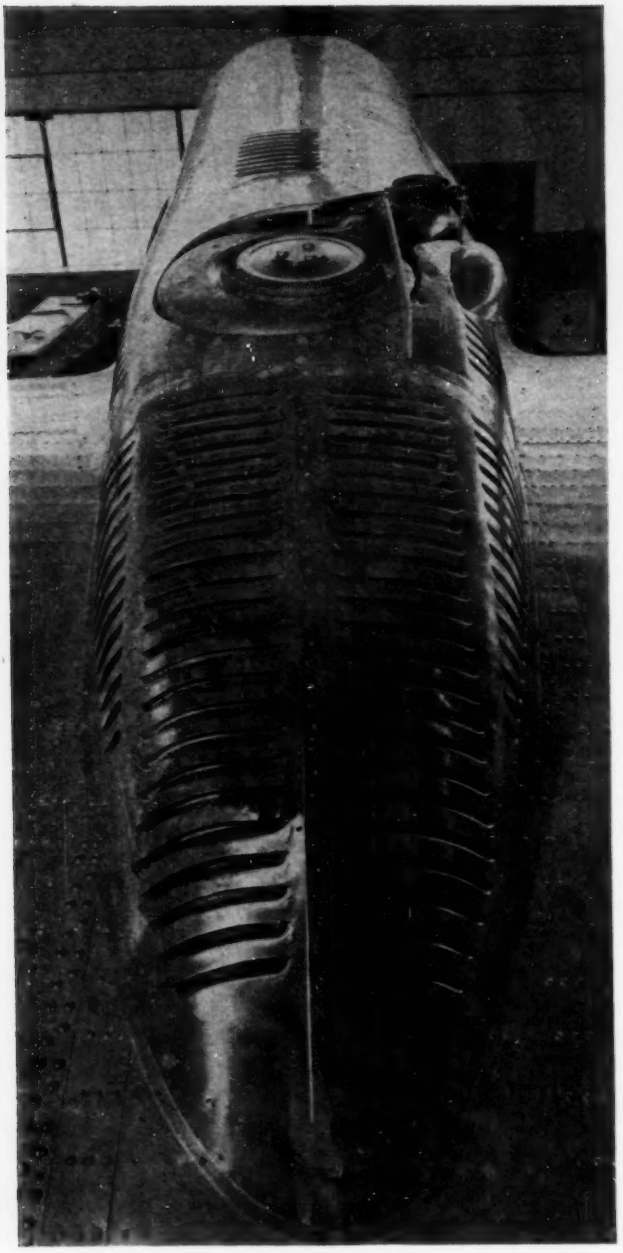
Awaiting the postwar period are the transport versions of the B-29. These transports will fall in the category of the long-range, high-altitude types. Time will be an item of ever-increasing importance. Altitude and power are the



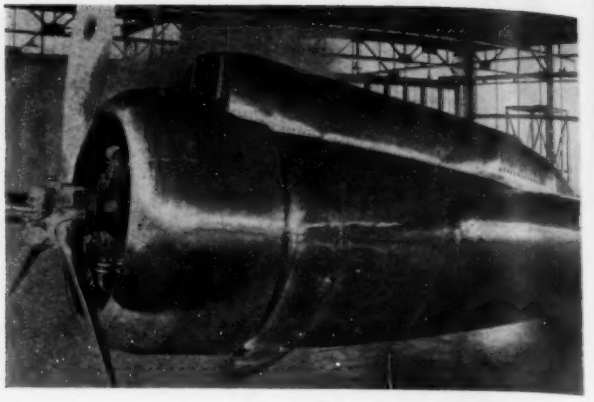
■ Fig. 1 – YIB-17A airplane

answer. Economical fuel consumption will enable large payloads to be carried, and the American public received the first glimpse of this type of transport in the record-breaking cross-country flight of the Boeing XC-97. Here the country was spanned in 6 hr and 4 min. The average ground speed was 383 mph, including a tail wind of 45 mph, and the flight altitude over the major portion of the trip was 30,000 ft. At this speed it was difficult to maintain radio contact or hold radio compass bearings with ground stations. Half-way across the continent the blackness of night in the east was visibly contrasted with daylight in the west, night and day all in one glimpse. The smoothness of flight is remarkable at this altitude and comfort is predominant, the cabin being pressurized to an equivalent of 8000 ft altitude. The cabin supercharging on the XC-97 is accomplished by bleeding the engine turbosuperchargers. Where necessary, slight amount of part throttle is employed to maintain sufficient turbo compressor discharge pressure, approximately 24 in. of Hg absolute to give 22 in. of Hg absolute in the cabin. Cabin pressure regulation is automatic. For commercial transport aircraft, thought has been given to a separate power package for electrical power generation, cabin supercharging, ventilation, and air conditioning. This will improve the engine powerplant installation and remove the burden of responsibility of functions other than the production of thrust. During glide, approach, landing, ground docking, and emergency conditions, the cabin and airplane functions will not be dependent on the engines.

At this point in the paper it is felt that an engineering tour across the drafting boards may be interesting, to present some of the design features of turbosupercharged powerplant installations, the main problems involved, and finally sketches of turbo installations for transport aircraft. Fig. 1 shows the Y1B-17A powerplant configuration. Note the turbosupercharger mounted on the top of the nacelle. The air for the turbosupercharger, intercooler, and heated air source enter at the front of the scoop, which is divided into three sections: louvers at the top and aft portion of the nacelle dump turbo and intercooler cooling air overboard. The turbo was unconventional in that the impeller entrance was located between nozzle box and compressor casing. In contrast with the highly efficient turbo lubrication systems of today, this turbo had a bulky external sump, and a series of grease cups for lubrication. The turbo rating was such as to supercharge a 700-900-bhp



■ Fig. 2 – Top view of top cowl installation



■ Fig. 3 – Side view of top cowl installation

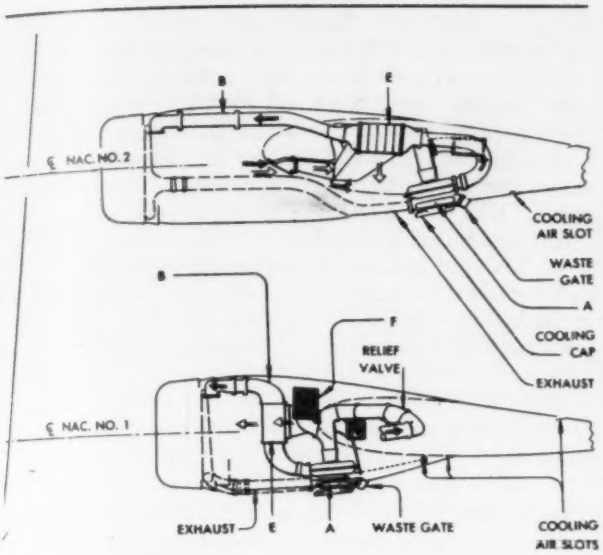
engine. This was the B-17's first experience with a turbo and flight tests showed excessive buffeting which was due to the relatively low velocity discharge cooling air and exhaust gas over the wing. Army requirements, at this time, stipulated that no exhaust gas be discharged below the wing. This was essentially for fire prevention and exhaust visibility from below. A feature of this arrangement is the high location of the turbo for fire safety, since fuel and oil will flow to the low point of the nacelle. The landing gear and tires are not affected by exhaust discharge during landing and take-off.

The disadvantages of this type of installation are poor intake scoop recovery during climb or at high angle of attack, low velocity discharge over wing. In addition, oil accumulations would be a hazard to maintenance personnel working on top of the wing. Turbo access and serviceability is also reduced with this type of top installation, which aerodynamically is not as satisfactory as low-slung nacelles which do not affect the airflow over the wing. The exhaust glare may also be annoying to pilot and passengers, and the turbo wheel is in direct line with the fuselage. This is a hazard in case of wheel failure.

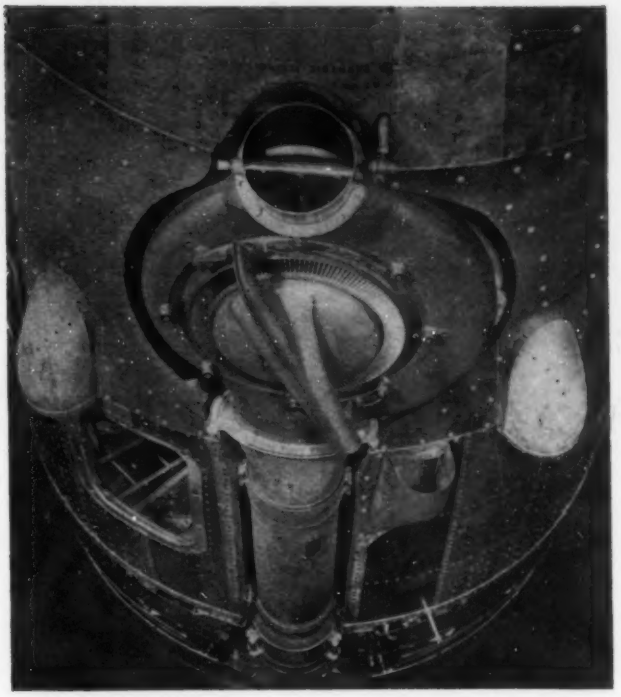
Figs. 2 and 3 show top and side views of this installation. This configuration was finally revised and a configuration which moved the turbo to the bottom part of the nacelle was evolved. On the inboard engines the turbos are located aft of the wheel well. The intake ducts start at the wing leading edge. The oil cooler is installed forward of the firewall in the nacelle. The intercoolers are located in the nacelles and wing. The cooling air discharge passes through the wing truss type of spars and exits at a series of slots just aft of the rear spar.

Fig. 4 shows the present B-17G nacelle arrangement which basically is the final Y1B-17A with minor revisions. Carburetor air filters, exhaust stack heater exchangers, cowl flaps, and so on, have been added. Improved turbo regulation and improved turbosupercharger maintenance and performance give the present B-17 excellent performance, the rated power critical altitude of the B-17G airplane being approximately 35,000 ft.

A bottom rear view of this turbo installation can be seen in Fig. 5. Note the waste gate, turbo cooling air scoops, and exhaust tail pipe shroud cooling air duct outlet. Also note the turbine wheel cooling cap installation. Generally speaking, the turbo installation fairs in very well with the nacelle contour and is easily serviced and removed. A dam fits completely around the upper portion of the turbo installation to prevent oil or fuel from coming in contact with the turbo nozzle box. Suitable drain holes discharge oil or fuel clear of the exhaust system. In addition, the accessory section is well ventilated. This increases the flash point of oil coming in contact with hot surfaces several hundred degrees. Oil contact-surfaces as high as 1100 F will not flash if adequate ventilation exists. A sketch of the present B-29 powerplant configuration may be seen in Fig. 6. In this installation the turbos are mounted as a dual installation parallel to sides of the nacelle. A main air duct or scoop is located on the bottom side of the nacelle providing air to the turbosuperchargers, intercoolers, and oil cooler. The turbos are conveniently located for service and maintenance and exhaust hoods for discharging the exhaust gases rearward are easily attached. The intercooler cooling air discharges over the top of the nacelle and the oil cooler air on the underside. The nacelle is of semi-monocoque construction and supports all the cowling,



■ Fig. 4 – Present B-17G nacelle arrangement – basically the final YIB-17A with minor revisions



■ Fig. 5 – Bottom rear view of turbo installation shown in Fig. 4

namely, nose, ring, and accessory, and differs from the B-17 construction which is an engine mounted cowl.

Fig. 7 shows the organs of the turbo system in more detail. Note the structure supporting the turbos. In addition, this structure strengthens the nacelle, since the semi-monocoque nacelle construction is not present in this region. The tail pipe shrouds, exhaust hoods, and turbo cooling air take-off duct can also be seen.

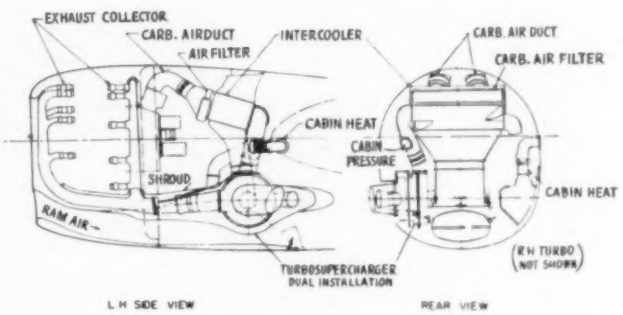
Fig. 8 is a rear view of the parallel turbo installation as tested on a B-17 airplane to demonstrate the feasibility of the dual turbo operation and the possibility of switching over from dual to single turbo for benefits in cruising. On the B-29 airplane the turbo speeds are within 500 to 1000 rpm of each other under all conditions of operation.

Fig. 9 shows a side view of the B-29 nacelle with turbo shroud covers off. Note the well which houses the exhaust pipe and turbo. Additional radiation shields protect the inner shroud. The waste gate extension can be seen just aft of the exhaust hood. Cooling air taken from the main intake duct is used for the turbo radiation cap and turbo bearings.

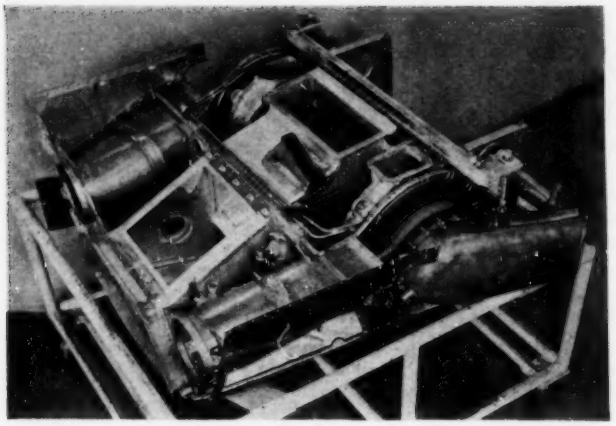
Fig. 10 is a side view of the nacelle with shroud covers installed. Note the clean appearance. The spring-loaded door shown in the forward part of the shroud cover provides ventilation for the turbo well.

Figs. 11, 12, 13, and 14 portray some ideas on powerplant arrangements which were evolved on the drafting boards. Fig. 11 is a novel arrangement and utilizes a bottom intake scoop located aft of the engine. This allows the engine cowling to be readily removed. A boundary layer bleed located at the top of, and slightly forward of, the scoop inlet will provide good pressure recovery. The typical crossing of induction system ducts which usually results when a cross-flow type of intercooler is employed is readily apparent. This should stimulate thought for parallel- or counter-flow types of intercoolers. Note the oil cooler, intake filter, and turbosupercharger locations. The rearward discharge from the turbo hood is extremely favorable for thrust consideration.

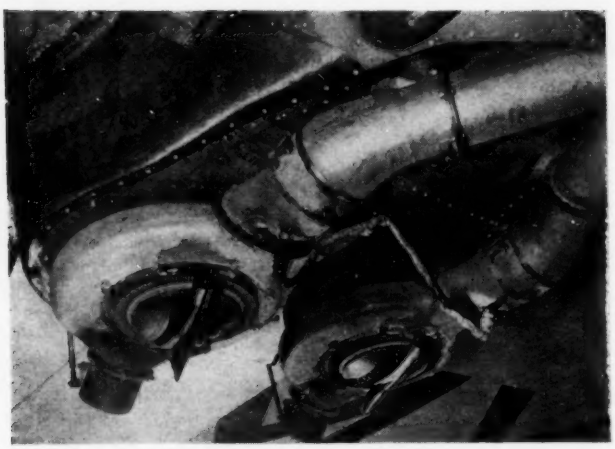
In this case, the turbo installation would have to be well dammed and shrouded to eliminate fire hazard, being located in the low point of the nacelle. A disadvantage also present with this arrangement is the exhaust discharge on the inboard landing gear and wheel at take-off, ground operation, and landing. Since the location of the main landing gear other than in the inboard nacelle body presents a problem, turbo locations on the bottom of the nacelle may be undesirable. Some aircraft have retracted the wheels into the wing proper. Fixing the nacelle firewall as close to the front spar as possible and keeping the wing structure free from powerplant items dictates a maximum overhung moment for a given engine. This results in a given nacelle length, and hence the available room behind the engine for the intercooler, turbo, and so on, will always be small. This results in the usual crowding problem. This space limitation makes it extremely difficult to juggle turbo, intercooler, preheat ducts, oil cooler, oil tanks, and so on, at will. In some cases an updraft carburetor would be advantageous. This reduces the duct path from the turbo compressor to carburetor and locates turbo, intercooler, and intercooler cooling exit air



■ Fig. 6 – B-29 inboard nacelle



■ Fig. 7 – Turbo system shown in detail



■ Fig. 8 – Rear view of parallel turbo installation as tested on B-17 airplane to demonstrate feasibility of dual turbo operation and possibility of switching over from dual to single turbo for benefits in cruising

ducts in the lower portion of the nacelle. Disadvantages would be the poor accessibility to the accessory compartment from under the nacelle but this could probably be accomplished by use of a large access door at the top. A down-draft carburetor engine would be extremely favorable for a top intake scoop design but this also has its shortcomings, as previously mentioned.

Fig. 12 is a schematic plan view of just the lower portion of a nacelle just aft of the scoop inlet. Here the oil cooler and turbo are located near the sides of the scoop body. Intercooler cooling air, engine charge air, and cooling air for exhaust shrouds, nacelle ventilation, and the like, enter in the central duct.

The turbo exhaust discharge is extremely favorable for jet thrust. Accessibility to both oil cooler and turbo are satisfactory in this case. The rather large expansion into the intercooler cooling passage may not be too efficient. Vaning and boundary layer bleed will tend to improve this condition.

Fig. 13 shows an arrangement quite similar to the others previously shown except that some exhaust system details are included. This combination employs a conventional exhaust collector with dual outlets, one on each side of the nacelle. A two-position valve, either open or closed, is installed at each exit. The turbo is strictly used for cruising economy and is bypassed for take-off, climb, and general high-power operation. Either a multispeed or variable speed internal supercharger will provide these high powers when required. When economical cruising is desired, the turbo and waste gate receive all of the exhaust gases. The waste gate flow is controlled by a valve at the waste gate exit to take full advantage of waste gate exhaust jet thrust, rather than a throttling valve upstream in the waste pipe. Thought has also been given to provide a variable outlet area for the main exhaust collectors when operating at the high powers where the turbo cannot be used. This gives maximum benefits of jet thrust. The design problems accompanying such an outlet valve are fairly evident. In one case the outlet flap measured 10 x 15 in. The high temperature and large gas forces create quite a design problem. A sliding door type of valve cannot be depended upon, due to possible seizing and binding.

With larger engines being studied, exhaust collectors are growing to greater proportions and the engine manufacturer must provide sufficient space behind the engine to allow for a suitable collector to be installed. Present difficulties are the lack of space due to the engine mount attachment locations and carburetor. This fixes the location of engine diaphragm or inner cowl and thus the allowable space for the collector.

The side view shows the turbo and oil cooler locations. In addition, the intake duct valving and carburetor air filter are also shown. Note that this particular study omitted an intercooler.

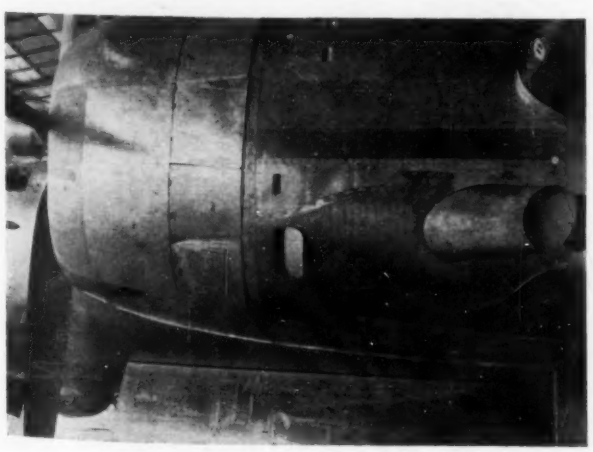
Fig. 14 shows a liquid-cooled engine installation incorporating a cruising turbo. The engine itself can be easily mounted to a simple cantilever type of beam structure much less complicated than a steel tubular type of mount structure. Accessibility is predominant with this type of mount. Dual oil coolers are used and are mounted on each side of the large glycol cooler. These dictate the intake

■ Fig. 9 – Side view of B-29 nacelle with turbo shroud covers off

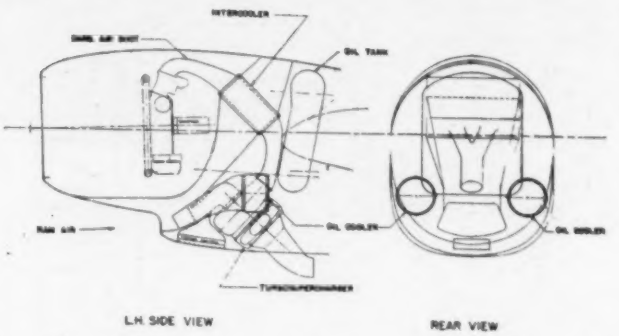
duct shape and depth. The coolers and turbo can be supported by the engine mount structure and are quickly detachable due to strap type of clamps and support brackets. All engine cowling forward of the firewall is readily detachable, thus allowing complete access to engine and accessories. In addition, the propeller was raised approximately 9-10 in. due to the present designs of liquid-cooled engines. This is extremely desirable from the ground clearance standpoint or possible increase in propeller diameter.

In general some of the items which have been gained from background and experience will be discussed with respect to turbosupercharger installations. For safety from fire, exhaust lines should be kept as near the skin location of the nacelle as possible. All shrouds should be adequately drained. Turbo installations, shrouds, and exhaust outlets should not be on the bottom of the nacelle. Locate oil tank, cooler, and oil and fuel lines on opposite side of nacelle from turbo installation. Positive nacelle ventilation should be provided for. With respect to safety from turbo bucket failures, turbos should be mounted such that the plane of the turbine wheel does not intersect central portions of the fuselage. At altitudes above 15,000 ft the cruising turbo should be very favorable from the carburetor ice prevention standpoint. The turbo may even be employed for the powers above maximum cruising for carburetor heat, since the turbo is less efficient in this region, being essentially designed for maximum cruise. The turbo system can contribute very little at low powers and altitudes, thus some definite means of carburetor preheat is desirable for carburetor ice removal. For carburetor ice prevention an alternate air intake is one method of providing additional protection. Sheltered air either from the accessory compartment or nacelle skate can be used during ice prevention operation. Thoughts have been given also to using air behind the engine on aircooled types. Here the engine cooling air experiences rises from 75 F to 150 F. Exhaust collector muffs can also be used but at the expense of maintenance, inspection, and possible excessive heat in the induction system due to collector failure.

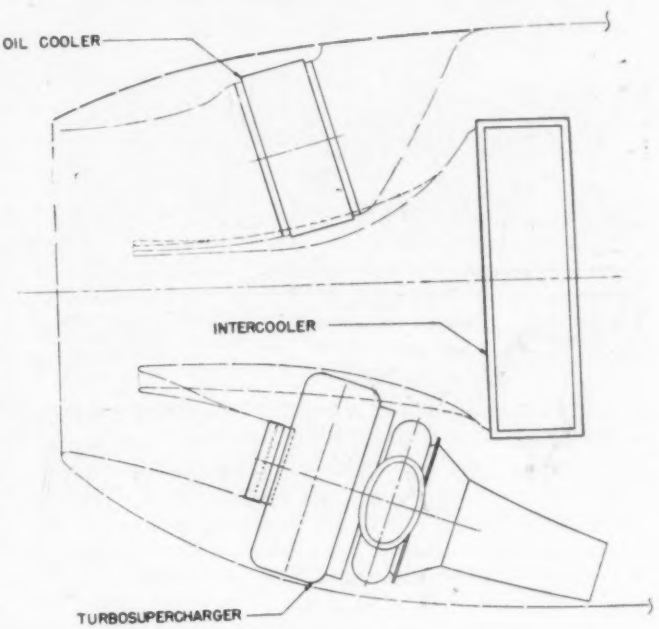
With respect to serviceability on turbosupercharged installations, there must be prompt access for inspection and cooling cap clearance check. Turbo removal should be a matter of minutes rather than hours. In this respect a three-point suspension is desirable, allowing the turbo to be slid into place. Turbo tolerances between compressor and nozzle box and waste pipe should definitely be improved for commercial application. Accessories should be easily removable, such as oil coolers, filters, and carburetors. Intercoolers, however, have not given trouble and need not fall in this category. Air filters should be installed and sealed by some type of compression fit. The sliding type of fit has not proved satisfactory due to binding and leakage. The steel-tube type of engine mount supports extending back to the fire wall is favorable and allows access from many points outside of the nacelle.



■ Fig. 10 – Side view of nacelle with shroud covers installed



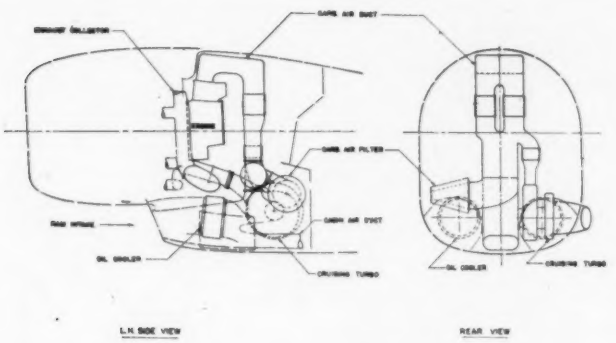
■ Fig. 11 – Powerplant arrangement that utilizes a bottom intake scoop located aft of engine



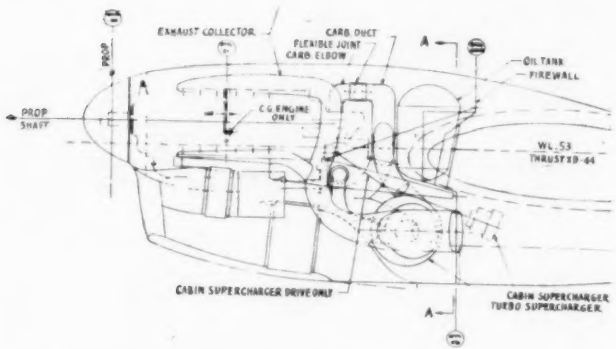
■ Fig. 12 – Schematic plan view of lower portion of a nacelle just aft of scoop inlet

In regard to functional characteristics of the turbosupercharger type of installation, as high a degree of ram as possible at the turbo inlet is desired. An inch of mercury ram at turbo compressor inlet may mean as much as 5000-7000 ft in engine critical altitude. If a filter is placed before the turbo in addition to poor ram at the filter face, the critical altitude as determined by limiting turbo speeds may be decreased as much as 8000-10,000 ft.

Turbo bypass ducts are desirable on full turbo installations, except where advanced rating requires use of turbo



■ Fig. 13 – Powerplant arrangement employing conventional exhaust collector with dual outlets



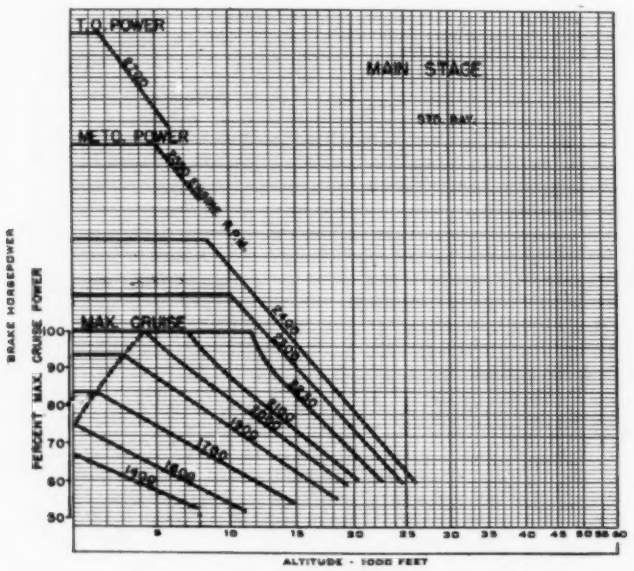
■ Fig. 14 – Liquid-cooled engine installation incorporating a cruising turbo

for take-off. This is desirable from the engine cooling and engine power standpoints. The oil tank is better located in the nacelle in interchangeable type of installations. If engine parts fail and metal chips are included in the oil circuit, the whole power package is easily removed. To reduce nacelle length, the central feeder type of oil system may have advantages, but the length of oil lines and possibilities of oil congealing are objectionable.

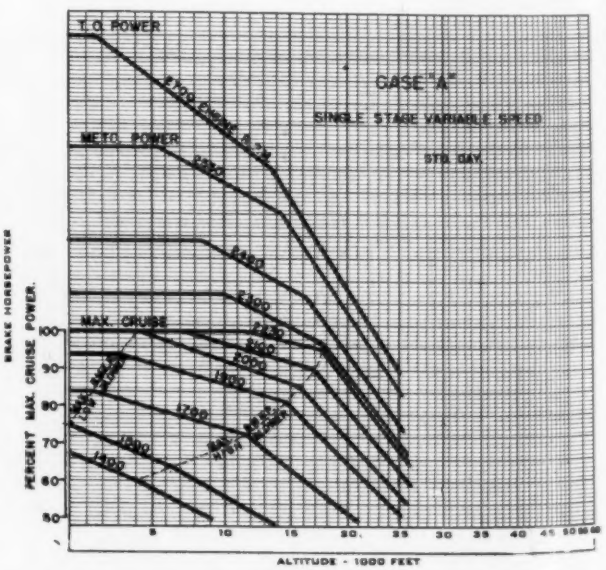
Of interest is the cooling drag which can be incurred with wasteful use of cooling air for accessories, shrouds, turbos, and the like. Careful thought should be given to this. On one installation alone approximately 130 lb per min of cooling air was used. This amounts to 38 thp or over 45 equivalent engine bhp per nacelle at the recommended cruising speed of the airplane or 3% increase in fuel consumption. In the design of exit areas and flaps for oil cooler, intercooler, and the like, sufficient area should be provided at small flap angles to provide sufficient cooling at design conditions. Flap angles above 8-12 deg usually cause buffeting in addition to the form drag of the extended flap. Good pressure recovery ahead of oil coolers and intercoolers is essential to give high static pressure aft of the coolers and hence higher exit velocities at reasonable flap angles. This keeps the cooling drag to a minimum.

At this point in the paper it was thought interesting to present the general process involved in arriving at the desired amount of supercharging and supercharging configuration for a 25,000-ft transport airplane. 25,000 ft was chosen since the range under consideration is over 3000 miles. In addition, the weather and riding comfort at this altitude are excellent. Since time is an ever important factor, it is desirable to fly at altitude. This of course means pressurized cabins, but past experience will make this problem relatively simple. Air conditioning, however, is still a big problem. Cabin supercharging without humidification and refrigeration in some cases will be uncomfortable for passengers. Cabin supercharging, which is entirely independent of engine operation, is desirable. This point has been clearly emphasized with experience gained on both the 307 Stratoliner and the B-29.

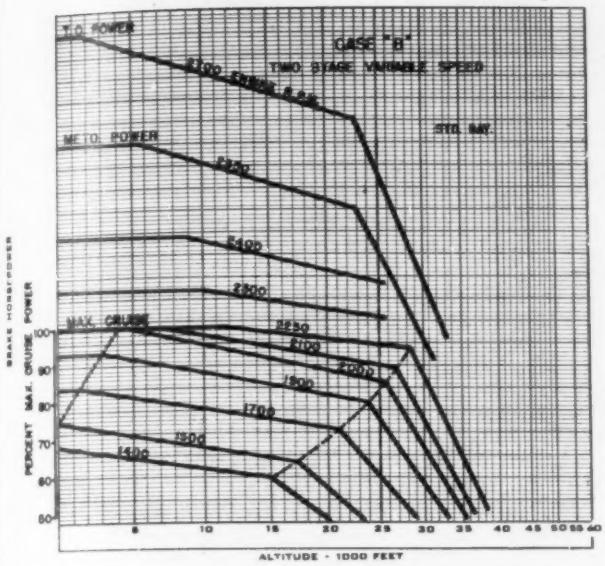
In the discussion which follows it is assumed that both basic engine and airplane are fixed and the problem is essentially one of choosing the most advantageous type of supercharging for 25,000-ft cruising. This might represent the case of a medium-altitude or high-altitude, high-performance military aircraft which may be considered for



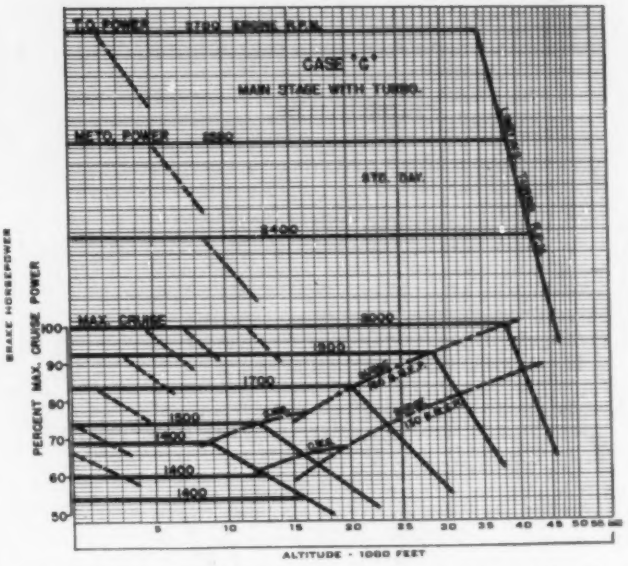
■ Fig. 15 – Basic engine – note that it falls extremely short of giving good altitude performance



■ Fig. 16 – Basic engine provided with increased internal supercharger ratios to improve altitude performance



■ Fig. 17 – Basic main-stage engine transformed into 25,000-ft unit

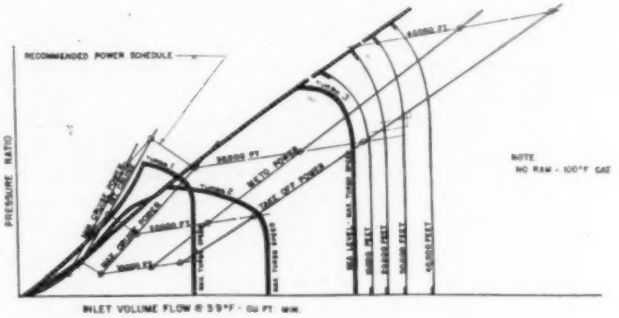


■ Fig. 18 – Basic engine with large turbo for all-out high-altitude-and-power performance with favorable cruising

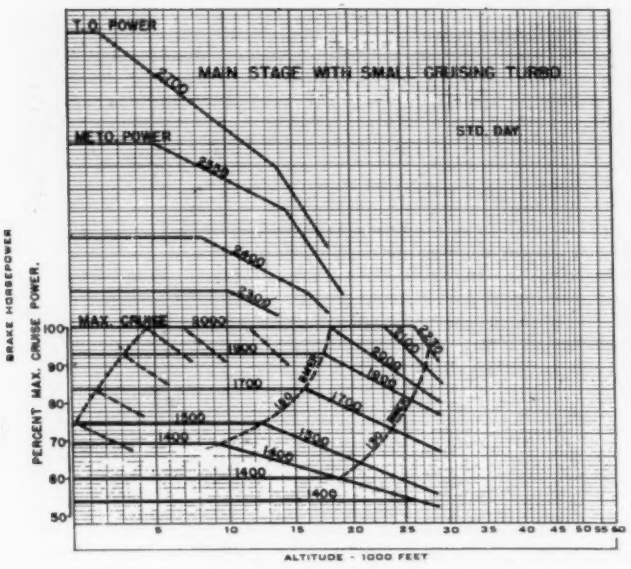
postwar transport application. The general power requirements with altitude and gross weight are approximately known after a study of desired cruising speeds and engine cooling requirements. The take-off power requirements and meto power requirements for one engine inoperative ceiling dictate the power performance of the engine. Thus, a basic engine size is fixed. Next, a study of several methods of supercharging to meet the power requirements is in order. Figs. 15 through 21 show a basic engine with several types of supercharging. Fig. 15 shows the basic engine, which falls extremely short of giving good altitude performance. This type of engine is representative of those used on domestic airlines. Take-off power is available up to 1500 ft and meto or maximum except take-off is available up to 5000 ft. Maximum cruising power can be obtained up to approximately 12,000 ft. It can be seen that this performance is somewhat marginal for take-off at high-altitude airports and rate of climb over 10,000-15,000-ft mountains. Where this type of terrain is frequently traveled, two-speed internal supercharging will provide the necessary power for rate of climb, and so on. However, the case of one engine failure on two-engine aircraft may necessitate emergency power operation of the remaining engine, reduction of flight speeds from optimum cruising, reduction in altitude, and so on, depending on the particular situation. The dotted line at the lower left portion of the chart shows the critical altitudes for maximum cruising bmep. (This is usually the highest bmep consistent with good engine life and long overhaul periods.) The horizontal lines to the left of the dotted line represent part-throttle operation at the maximum cruising bmep. Thus, it is seen that maximum cruising bmep can only be obtained below approximately 4000 ft. For a given power, operation to the right of the dotted line represents higher engine rpm and hence increased specific fuel consumptions. This is due to higher engine friction horsepower and increased engine charge airflow requirements. The curved portion of the 2230-rpm line shows that fuel enrichment is present. This enrichment is mainly for purposes of engine cooling and usually starts at maximum cruising powers and continues up to take-off power. This enrichment is more pronounced on aircooled engines than on liquid cooled. Fuel-air ratios of 0.065 to 0.070 are provided by the carburetor for the cruising range and these increase up to 0.11 for take-off powers.

To improve the altitude situation, increased internal supercharging, as shown in Fig. 16, can be used. The basic engine shown just previously is provided with increased internal supercharger ratios.

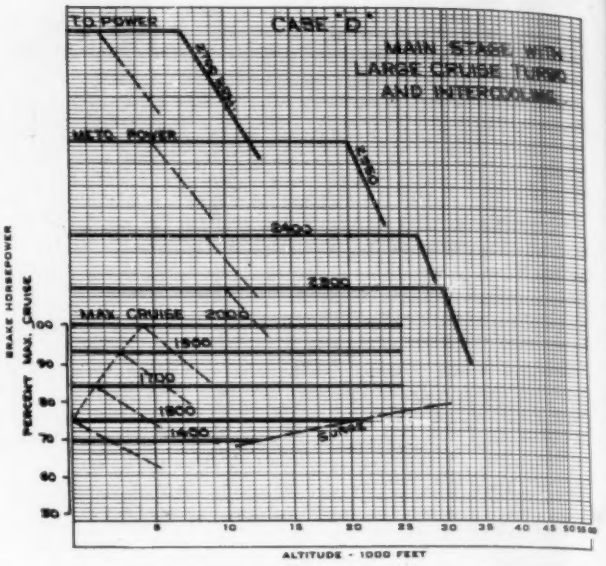
As altitude is increased, the power drop off at constant rpm is dictated by limiting intake pipe temperatures. The region included between the two dotted lines represents the cruising range and represents 20 to 30 lower bmep at the high ratio line. This reduction is based partially on the bmep absorbed by the additional supercharging. At the lower rpm's the effect of this supercharging is not much better than the basic main-stage engine since the supercharger pressure rise is a function of engine speed. This engine will give excellent 15,000-ft performance. A fairly large percentage of sea level meto power is still available at 15,000 ft. This engine would serve very well



■ Fig. 19 – Curves illustrating proper method of turbo selection



■ Fig. 20 – Basic engine provided with turbo No. 1



■ Fig. 21 – Basic engine provided with large cruising turbo

in routes where topographical conditions necessitate 15,000-ft operation over a main portion of the route.

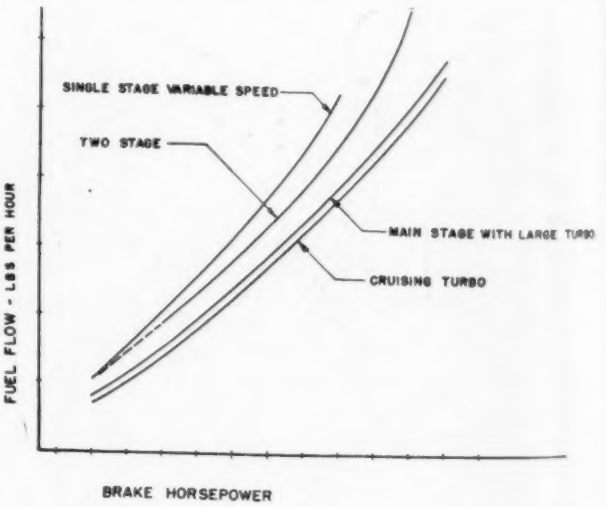
Specific fuel consumptions at equal power will of necessity be higher at maximum high ratio due to both increased engine rpm and increased internal supercharger horsepower requirements.

To carry the engine geared type of supercharging one step further, an additional supercharger can be built into or attached to the rear of the basic engine. This supercharger can be of the multiple-speed or variable-speed type. The latter utilized a fluid coupling type of drive. The basic main-stage engine now is transformed into a 25,000-ft unit, as shown in Fig. 17. This type of engine is best suited for a 25,000-ft type of military airplane. It does not possess the high-altitude performance of this same engine fitted with a suitable turbosupercharger. It is true that jet stacks employed with this engine would show large benefits at high altitudes and high speeds, but the fuel economy is almost certain to be inferior. Also, the noise produced by jet stacks is objectionable. In addition, with a turbo system considerable jet thrust is available from the waste gate and turbo hood if variable exit areas are provided for these. At the lower power and cruising speeds jet thrust contributes less due to slower speeds. Suitable intercooling is employed with this type of engine to maintain carburetor air temperature limits set by the engine manufacturer. For a commercial high-altitude transport excesssive supercharging exists at take-off and meto power conditions and this weight penalty would be undesirable. The high-altitude, high-power design features could be removed, however, from this type of supercharger to give only the cruising power region included between the two dotted lines. Sufficient supercharging, however, could be included for meto at 15,000 to 20,000 ft. The large flow range required to meet both cruising and meto power, however, sacrifices supercharger efficiency at some operation. It is probably better to design the supercharger for maximum cruise operation and take sacrifices at the higher powers.

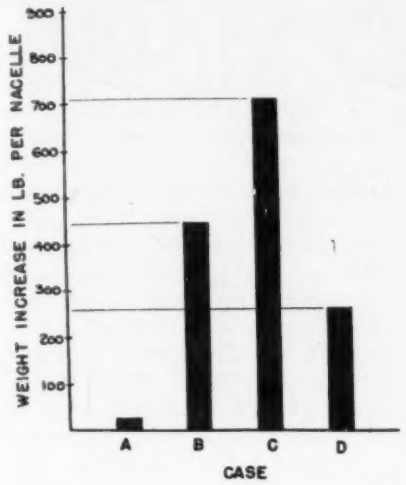
For purposes of interest the basic engine is fitted with a large turbo specifically for all-out high-altitude-and-power performance, with favorable cruising in addition. This is shown in Fig. 18.

Here military or take-off power is maintained to 35,000 ft and maximum cruise to 40,000 ft. Note the limitations on the lower cruise powers, namely closed waste gate and compressor surge. This indicates the large engine airflow flow range which the compressor and turbine are trying to meet. Bmep must be reduced, accordingly, in the cruising range to avoid these limitations with some penalty in specific fuel consumptions. The weight of this installation would exceed any of the others previously discussed due to the large turbo or turbos required, the large intercooler core size, and the larger and stronger exhaust systems for the high-altitude conditions.

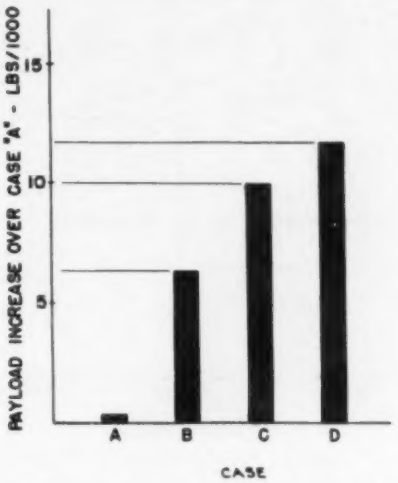
Now to formulate some concepts for a suitable turbosupercharger to provide maximum cruising benefits at, say, a design altitude of 25,000 ft. This altitude is chosen for several purposes, namely speed and operation free from



■ Fig. 22 – Fuel-flow comparison – 25,000 ft



■ Fig. 23 – Nacelle weight comparison



■ Fig. 25 – Effect of turbo installations at 25,000-ft altitude and optimum cruising speed of 295 mph on payload increase for 3500-mile flight

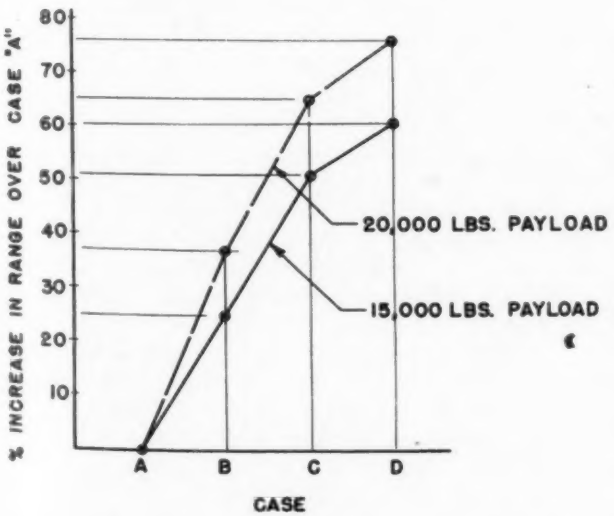
undesirable weather conditions, such as icing and head winds. This means schedules can be met without aircraft being grounded due to unfavorable weather conditions enroute. In addition, flight at this altitude provides the optimum in passenger comfort.

Fig. 19 illustrates a method of proper turbo selection. The basic coordinates of the chart are pressure ratio versus volume flow of the turbo compressor. The envelopes of three different sizes of turbo are shown in heavy lines. The limits of operation of the compressor as shown by the left boundary is surge or pulsation. This is a condition of too high a rotational speed of the compressor for the volume flow passing through the compressor. This is similar to stalling of an airplane wing at too high an angle of attack. The right-hand boundary becomes vertical at the higher flows and shows the maximum turbo speed allowed by the turbo manufacturer. Superimposed on these coordinates are engine airflow and engine carburetor deck pressure requirements for various powers and altitudes. The engine power requirements are based on some chosen schedule such as minimum engine rpm, constant bmep for cruise, and propeller load up to rated power. Thus, if all-out maximum cruise performance is desired, turbo No. 1 is most favorable. If the best turbo for obtaining a meto power ceiling of 25,000 is desired, turbo No. 2 is most favorable. To obtain a 35,000-ft meto ceiling, turbo No. 3 is most desirable.

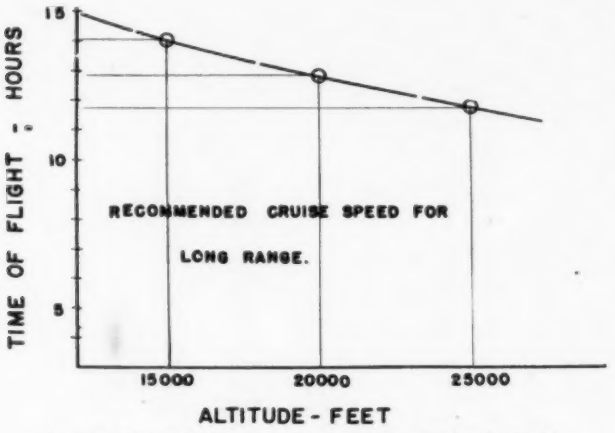
If all-out military performance at 40,000 ft is desired, none of the turbos shown will satisfy the requirement and an even larger turbo will be required. The parallel lines shown to the right of turbo No. 3 show how the limiting turbo speeds shift to the right as altitude is increased. This is due to the fact that the abscissa is in terms of inlet volume flow at sea-level condition. Actually, supercharger performance is a function of the Mach number at the compressor inlet, and the compressor speed. Thus, at the lower inlet temperatures at altitude, the limiting turbo speeds are reached at some higher equivalent inlet volume flow than that corresponding to limiting speed at sea level. The short dashed lines shown on the take-off or military power line is a method of determining critical altitude by the simple process of diminishing intersections. When the horizontal altitude lines intersect the limiting turbo speed altitude lines at the power required, the critical altitude is determined. Thus, turbo No. 1 is chosen, say to provide optimum cruising performance by the method of selection of this chart.

Fig. 20 shows the resultant performance of this turbo with the basic engine for standard-day conditions. No intercooling was assumed and a maximum carburetor air temperature which might be employed without seriously affecting engine life was chosen. At 25,000 ft this performance shows fuel economy considerably improved over the case of the single-stage, multispeed engine which was shown previously in Fig. 16.

Although the high-power performance as shown by the meto power line does not apparently give sufficient power



■ Fig. 24 – Range comparison for 25,000-ft altitude – fixed payload, 295 mph



■ Fig. 26 – Relative hours to complete 3500-mile flight

at 20,000 ft for one or more engine inoperative ceilings, the engine manufacturer could improve this considerably by including more internal supercharging in the variable-speed supercharger; this means an increased high gear ratio.

Another arrangement would be to use just the basic engine with a larger turbo than the one just previously discussed. This would give 20,000-ft meto performance and possible 7000-10,000-ft take-off powers at high airports, but would involve an additional weight penalty and a more difficult problem in fitting the turbo into the nacelle. Fig. 21 shows this combination (case *D*).

Fig. 22 shows the relative fuel consumptions at 25,000 ft with the various configurations just discussed. Note that the cruising turbosupercharger (case *D* of Fig. 21) shows the best performance.

Fig. 23 shows the relative nacelle weight increase with the various configurations discussed. Note the 700-lb weight increase per nacelle for case *C* which was the high-power, high-altitude turbo installation.

A brief range study was made for the various configurations.

The same airplane polar was assumed in all cases. This assumption is not exactly true since slight differences in nacelle weights and cooling drag will exist, but these will not be as important as the basic differences in fuel economy. Nacelle weight differences, however, are included in the final range computation since landing weights will be different, assuming equal payload. Weight empty conditions will differ only by the relative nacelle weight differences. In addition, gross weights at take-off are not the same, but are based on the maximum power available at 25,000 ft to fly this gross weight at optimum cruising speeds.

Fig. 24 shows the increase in range over case *A*, which was the single-stage, variable-speed engine. This range comparison is for 25,000 ft, fixed payload, and optimum cruising speed of 295 mph. Note the 76% increase of case *D* over case *A* and the increase of case *D* over both *B* and *C*. Although the range is actually greater for the 15,000-lb payload condition, the 20,000-lb payload gives the greatest relative increase. This shows that with higher gross weights and hence higher engine powers required, the turbo installations provide increased benefits.

Fig. 25 illustrates the effect of the turbo installations at 25,000 ft and optimum cruising speed of 295 mph on payload increase over case *A* for a 3500-mile flight. Case *D* shows approximately 12,000-lb increase in payload over case *A*. Since time is an extremely important item, and it appears that it will be even more so in the future, flight at altitude is desirable. Fig. 26 shows the relative hours to complete a 3500-mile flight. At approximately 12,000 ft, this would require 14.8 hr, at 25,000 ft 11.8 hr, and at 30,000 ft slightly more than 10.8 hr. If speed rather than payload is important, a 3500-mile flight could be accomplished in approximately 9.9 hr at 35,000 ft.

In the above range and payload analysis, assumptions of equal airplane total drag were used. In addition, the effects of jet stacks on cases *A* and *B* were not presented, mainly due to the attendant unsatisfactory noise level and the problems of properly soundproofing the airplane for these conditions. Also, jet effects show up more pronounced at high powers and altitudes. Since only maximum cruising is considered at altitudes, the jet thrust effects will be smaller. A satisfactory jet stack installation to discharge directly rearward is not a simple problem. Jet stacks used as engine cooling air augmentors may be extremely advantageous in installations where engine cooling is inherently poor. Even with these effects included in cases *A* and *C*, namely the single-stage, multispeed and two-stage, multispeed engines, it is almost certain that the turbo would pay better dividends, due to its flexibility of operation and the saving of power which is wasted in a hydraulic type of drive coupling when slip occurs or in a multispeed geared supercharger, where it is necessary to throttle each time a higher speed ratio is utilized.

Summarizing the apparent favorable features of turbosuperchargers as applied to postwar transports, the following items are pertinent:

1. Increased fuel economy due to recovery of waste exhaust heat energy, which normally is inefficiently used and apparently is difficult to utilize in jet stacks due to high noise level.

2. The turbo can be specifically designed for a relatively narrow range of airflow, mainly the cruising power range. High compressor adiabatic temperature efficiencies may possibly be realized.

3. Turbo performance is relatively independent of engine speed, hence greater flexibility in supercharging and simplification of engine operational procedures are realized.

4. Turbo life and reliability are definitely good, as indicated from past experience, and the control systems are well proved, positive, and responsive.

5. The turbo, if designed specifically for cruising operation, is a compact unit of relatively small dimensions and is easily serviced and maintained.

6. Turbo systems in general have remarkable acoustic filter effects, as evidenced on both B-17 and B-29 airplanes. This will add much to passenger comfort during cruising operation.

7. The carburetor preheat characteristics of the turbosupercharger at altitude will eliminate carburetor icing hazards. For lower altitudes for both high and low power preheated air taken from a sheltered source will, however, be necessary.

Thus, the immediate future may see the cruising turbo pay dividends on long-range, high-altitude flight. This device represents the first step in improving the overall thermal efficiency of the reciprocating engine type of powerplant.

The MOTOR COACH of the FUTURE

by L. H. SMITH

Vice-President in Charge of Engineering
General American Aerocoach Co.

BELIEVING no attempt should be made to prophesy, without first delving into the past to see what has happened, your author briefly reviews the progress of the motor-coach industry over the past 20 years.

He has selected 1925 as the beginning of our modern period of progress, since it was about this time the 25-passenger intercity coach was introduced. At that time there were approximately 2500 intercity and 11,000 city coaches in operation in the U. S. In 1944 these figures were reported to have increased to 30,000 intercity and 51,955 transit coaches, of which 3555 of the latter are trolley coaches.

In the opinion of your author the 1944 estimates are much too high because in the past 10 years the manufacture of motor coaches in the United States has been as shown in Table 1.

Your author believes it reasonable to assume that the total number of coaches in operation at any given time will not, to any great degree, exceed the total production of the preceding 10 years.

In 1925 the average intercity fare was 3.22¢ per mile and the average operating cost was 15.09¢ per bus-mile (including depreciation, fuel, maintenance, oil, and wages).

In 1927 the 33-passenger intercity bus was introduced and the average passenger fare had dropped to 3.1¢ per mile.

Due to the high production of this vehicle, its relatively higher carrying capacity, growing public acceptance, and the steady improvement in design and construction, the average passenger fare rapidly continued to decline.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 8, 1946.]

In 1935 the rate of fare had dropped to 2.54¢ per passenger-mile while the operating cost per bus-mile had risen to 19.6¢. The increase in operating cost per mile was due, largely, to higher wages, heavier vehicles of larger seating capacity, more advertising, and *higher taxes*.

About this time (1935) the 37-passenger intercity bus was introduced and passenger fares continued to tumble, reaching a low of 1.55¢ per mile in 1941.

The operating cost in the meantime continued to rise and in 1940 was 22.03¢ per bus-mile and 27.72¢ in 1943.

It should be noted that the above figures do not include any trolley coaches, gas-electric trolleys (all purpose), nor strictly suburban type of operations. Furthermore it is impossible to tabulate city fares on a mileage basis, because of the flat fare system, hence no transit type of vehicle is included.

The average passenger fare per mile from 1925 to 1944 is shown in Fig. 1. The figures along the top represent the intersecting point of the curve and the date, and are expressed in cents.

The average operating cost per bus-mile from 1925 to 1944 is shown in Fig. 2.

Fig. 3 shows Fig. 2 superimposed on Fig. 1 on the same scale. The first marginal figures to the right indicate the average fare per mile when compared with the descending curve. The second marginal figures indicate the average operating cost when compared with the ascending curve.

You may well ask how the industry has been able to reduce fares steadily in the face of such drastic increases in operating costs. This is rather a difficult question to answer.

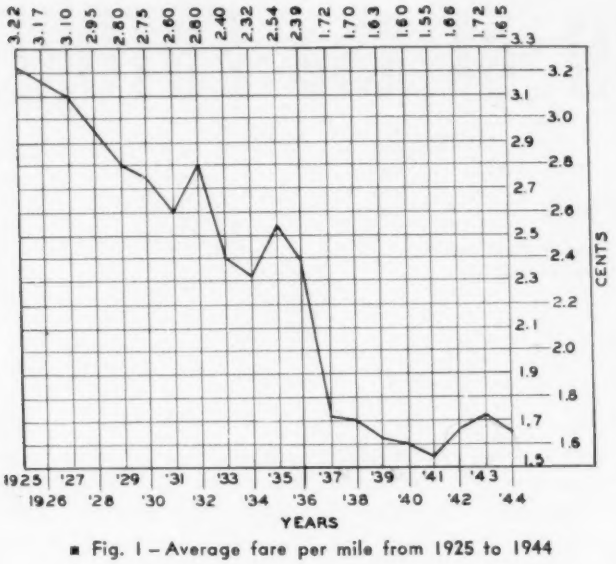
This seeming feat of magic is *partially* explained, how-

AFTER reviewing the progress that has been made in the motor-coach industry in the past 20 years, the author projects this progress 20 years into the future.

He predicts, for instance, that in this period long-distance buses will be 102 in. wide and 40 ft long, because he feels that this space will be needed by the designer to build into these buses the comfort so essential to the continued growth of the industry.

In addition, he makes many specific predictions as to what he thinks the future holds in the way of safety, comfort, eye appeal, and public service for the buses of the next 20 years.

THE AUTHOR: L. H. SMITH (M '37) is vice-president in charge of engineering of General American Aerocoach Co. A veteran of World War I and graduate from the University of Michigan, he has specialized in automotive engineering, having been on the staffs of Cadillac Motor Car Co., Chrysler Corp. and Yellow Truck & Coach Mfg. Co.



■ Fig. 1 – Average fare per mile from 1925 to 1944

ever, by the increase in carrying capacity of the average bus over the same period, 1925 to 1944. The average carrying capacity of intercity motor coaches from 1925 to 1944 is shown in Fig. 4.

During the past two decades the public has become increasingly bus conscious. Buses have been accepted as a means of economical transportation by millions of people, who ride them daily.

Table 1 – Production of Motor Coaches in U. S.

Year	Intercity	City	Trolley
1935	2255	3806	211
1936	2610	4572	538
1937	2430	3908	462
1938	2167	2498	184
1939	2415	3918	587
1940	2001	3984	310
1941	2088	5600	411
1942	3968	7200	386
1943	1691	1251	117
1944	1927	3807	55
Total	23,552[a]	40,544[b]	3241[b]

[a] National Association of Motor Bus Operators "Bus Facts."
American Transit Association "1945 Transit Fact Book."

We in the bus industry rather resent the inference sometimes expressed by the statement, "People ride buses only because they are cheap."

We do not believe this to be true. In our opinion people ride buses for many other reasons, such as:

Frequency of service.

Ability to board the bus in close proximity to home.

Delivery into the heart of town.

Safe, comfortable ride.

As a result of much advertising, more and more people are availing themselves of the long-distance bus for vacation trips.

This increase in patronage is most surely responsible for a major portion of the decrease in fares. In 1925 it was necessary to carry only 4.69 passengers to pay the operating cost of the bus. In 1944 it was necessary to carry 17.17 passengers to achieve the same result. This is shown clearly in Fig. 5.

During this period the average passenger capacity of intercity coaches had increased from 21 in 1925 to 34.1 in 1944. This increase, however, has failed to keep pace with the reduction in fares.

This is proved in Fig. 6, wherein is shown the per cent of average capacity necessary to pay operating cost of bus by year from 1925 to 1944.

You may now wonder, "What has all this to do with the motor coach of the future? This is ancient history and a matter of records." Your author replies: "Correct you are, but it took a whale of a lot of time and digging to get it." (At this time let me express my appreciation to *Bus Transportation*, the Interstate Commerce Commission, the American Transit Association, and the National Association of Motor Bus Operators for assistance rendered in compiling the statistics from which the foregoing charts were designed.)

Now let us project just one curve 20 years into the future. For this peek we will select Fig. 4 to see where the curve projection method will lead us. Fig. 7 shows Fig. 4 so projected. The curve 1925 to 1944 is actual and the dotted curve 1944 to 1964 is projected.

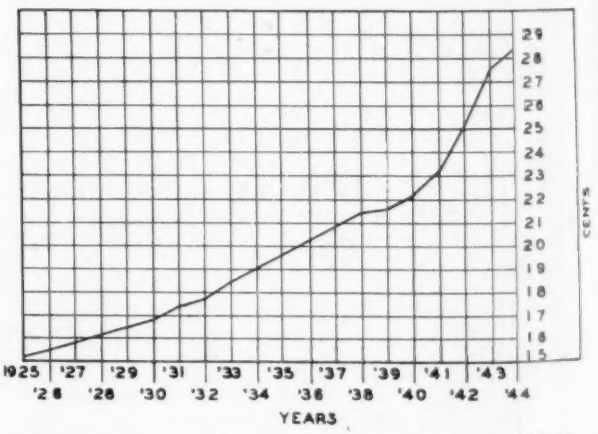
Obviously this method of prophesying may lead to trouble. For instance, were we to project this curve to the year 2005 we would find ourselves riding in a 74-passenger vehicle. This, while entirely within the realm of possibility, we believe to be entirely impractical.

Note: Speaking strictly as a designing engineer, I should like to go on record at this time, to the effect that the design of such a vehicle would present no insurmountable obstacles.

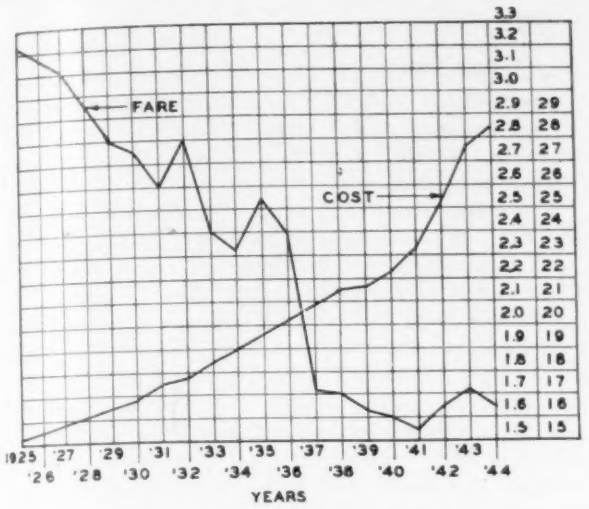
We are, therefore, going to assume the prerogative of an author and dispense with all scientific methods of prognostication. We are going to attempt to paint a word picture of what we believe to be a *practical* "motor coach of the future" and then explain, as clearly as space will permit, the reasons for our belief.

Note to my competitors: The bus described hereinafter is *not* on our drafting boards at this writing.

The motor coach of the future will be longer and wider than at present. How much will depend upon governing regulations and the progress made by the highway engineers in constructing the type of highways we shall need.



■ Fig. 2 – Average intercity bus operating cost per mile from 1925 to 1944



■ Fig. 3 – Fig. 2 superimposed on Fig. 1

I think it safe to assume, however, that the long-distance bus will be 102 in. wide and 40 ft long. In this space the designer can build the comfort so essential to the continued growth of the industry. And growth of the industry we must have. The bus business is big business as witnessed by the following figures:

In 1944 buses carried 19,082,900,000 passengers exclusive of school buses and sightseeing trips.

In 1943, the federal automobile taxes amounted to the staggering sum of $313,800,000.

In 1944 transit coaches alone operated a total of 1,845,600,000 miles.

In 1944 the payroll of the transit industry alone was $599,000,000, or an average earning of $2,475.

The total investment of motor-coach operators runs into a staggering sum.

Now to get back to our motor coach of the future; 41 passengers can be seated in 19-in. wide chairs on 40-in. centers.

Probably, a ladies' and also a gentlemen's retiring room will be provided. This presents no particular problem.

Present-day gasoline and diesel engines will be obsolete. The power generating equipment may consist of a gas turbine connected to a small, high-speed, glass-insulated, d-c generator. Four small, high-speed electric motors of perhaps 75 hp each may be used to provide flexibility in the final drive.

This type of drive will develop high speeds economically, and provide constant, high torque. Speeds in excess of 100 mph *on suitable highways* will be accepted as common, and acceleration will be at the rate of 3½ mph per sec.

The development of this type of power will become practical only when our friends the metallurgists give us turbine wheel material and a toyer which will stand up under intake temperatures of 2000 to 2500 F. Then and then only will the efficiency of a gas turbine become such as to make it practical in the 400-500-hp range.

The efficiency which we may expect from such a turbine is shown in Fig. 8.

Fuels may consist of two nonvolatile liquids which, when mixed in the precombustion chamber, will unite to form a combustible material which will burn at temperatures above 2000 F. All fire hazards, however, will be minimized because of the nonvolatile, noncombustible characteristics of the separate ingredients.

There is a possibility that atomic energy may become the means of generating heat and thus securing power. We believe a few words about this subject may be of interest.

In 1905 Albert Einstein clearly stated mass and energy were equal. He concluded that the amount of energy E equivalent to a mass M was expressed by the equation $E = M C^2$ where C is the velocity of light. Science was unable to prove this theory for nearly 40 years. It has since been shown to be absolutely correct by the scientists working in the technological laboratory of the Clinton Engineering Works of the Manhattan District at Oak Ridge, Tenn., and elsewhere.

Transposing the above equation into figures we find that one pound of mass contains 186,000 x 5280 = 982,080,000 ft-sec or $982{,}080{,}000^2$ = 964,481 trillion lb-ft.

This amount of energy is beyond the comprehension of the average human mind.

We have learned how to release this energy. At this writing, to the best of your author's knowledge, however, we have not learned how to release it in small quantities of less than 2½ lb nor in a manner other than instantaneous. This method is disastrous as the Japs can testify: at least those still alive.

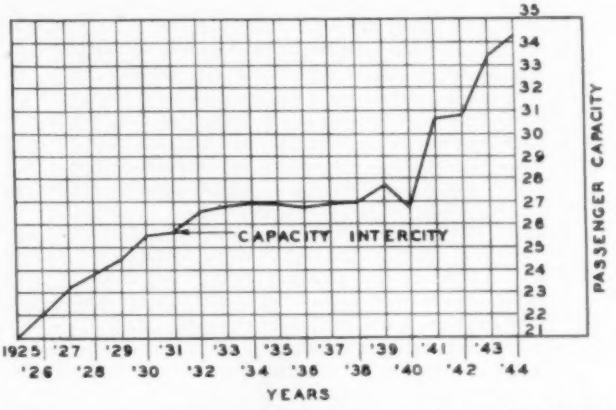
Once we learn how to control the release so it can be utilized as needed we will become possessed of a source of power abundant beyond our wildest dreams.

Our information on this subject may even now be obsolete, as it is reliably reported one of our largest electrical manufacturing companies has constructed or is in the process of constructing an atomic-powered railway locomotive.

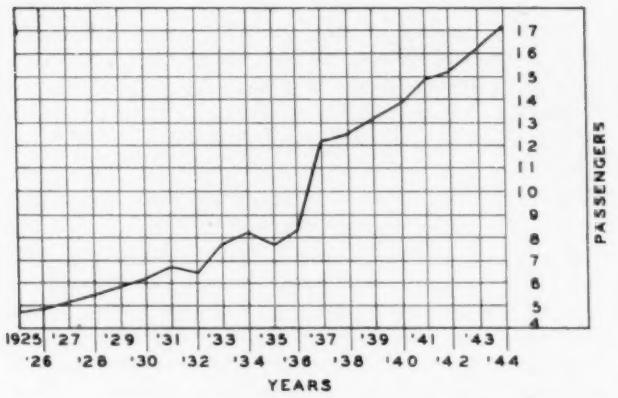
To keep pace with the higher speeds many new developments will be necessary.

New tires are required, of smaller diameter, larger cross-section, and employing no air. Thus, punctures and blowouts will be rendered impossible.

Because of the smaller wheels the brakes will move away from the wheels. Electric brakes will be developed or the polarity of the motors may be reversed for this purpose. The secondary braking system may be hydrauli-



■ Fig. 4 – Average intercity bus seating capacity from 1925 to 1944



■ Fig. 5 – Number of passengers necessary to pay operating cost from 1925 to 1944

cally actuated. Eddy current and water retarders offer other possibilities.

Steering will be of the power type, electric and/or hydraulic.

The driver's only controls will be the throttle and brake.

Doors will be hydraulically actuated, as will windshield cleaning devices.

Defrosting on all glass surfaces will be accomplished electrically.

Compressed air will disappear from the vehicle.

An entirely new suspension system will be employed. All leaf springs will be eliminated. Suspension will be at or near the c.g., making the bus much more stable.

All passenger compartment windows will be double glazed and the body completely soundproofed.

All interiors will be completely air conditioned, and we mean air conditioned, not just cooled. Twenty to twenty-five cubic feet of fresh air will be introduced per seat per minute. This air may contain as much as 132 grains of moisture per lb. This must be reduced to at least 72 grains. This, together with the sensible temperature reduction necessary to human comfort, the load imposed by the passengers, and so on, will require about 10 tons of equivalent refrigeration. All baggage compartments will be air conditioned.

This will be provided by exhaust heat, using a vapor system similar to the one now employed by the railroads. Turbine exhaust heat will be ample for this purpose.

Each passenger will have individual draft control similar to that provided in modern passenger transport planes.

The inside of the bus will be pressurized.

This improvement will prevent the infiltration of dust and dirt. All incoming fresh air will be filtered.

All maintenance will be simplified. All electrical connections will be of the push-pull type. All liquid lines will have a quick disconnector and sealed joints. Powerplants will be removable in 10 min. Axles will be removable in less. All joints and bearings of the control system will be of the greaseless type. Seat backs and cushions will be readily removable and bound in canvas. The upholstery will be of the envelope type so that it can be unzipped and cleaned or replaced in a matter of minutes without removing the seat.

The driver will have a public address system for communicating information to the passengers.

There will be a multicircuit radio telephone switchboard, entirely automatic. Any passenger may pick up a phone, without leaving his seat, call any telephone number in the country and carry on a conversation while traveling at more than 100 mph.

A multichannel radio receiving set will be installed. Headphones will be built into the headrest of each seat. Selection of programs will be by means of an individual switch.

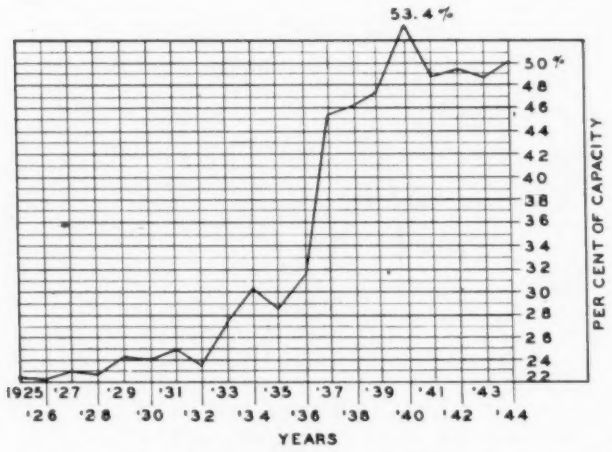
Circulating ice-cold water will be available at each seat.

Individual windows will be round with polaroid glass. The inner frame will be rotatable to permit total or partial blackout as the passenger desires.

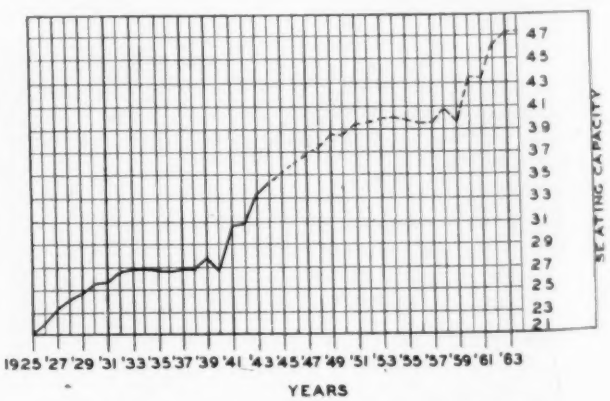
All windows will be emergency exits. Passengers will be able to kick out the entire framework in case of emergency.

All headlights and windshields will have polaroid glass. The poles will be set at a 45-deg angle and all slanting in the same direction. This will nearly black out oncoming headlights but will in no way disturb normal vision. The driver's own lights will appear normal and will clearly show him all objects as at present.

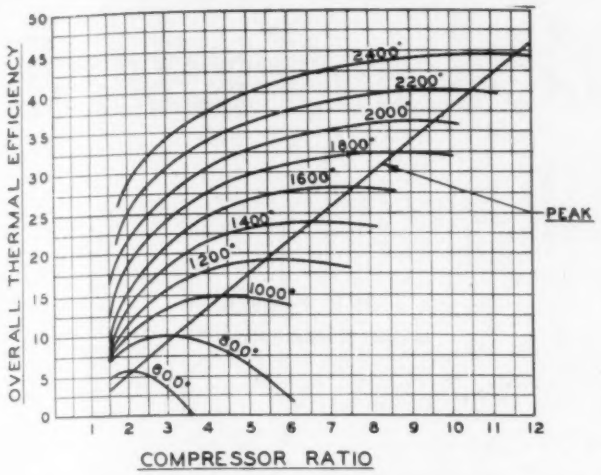
Fig. 9 represents a driver's view of an oncoming coach. Note that the direction of pole lines is the same on both



■ Fig. 6 – Per cent of average capacity necessary to pay operating cost of bus from 1925 to 1944



■ Fig. 7 – Fig. 4 projected 20 years into the future



■ Fig. 8 – Variation of overall thermal efficiency of turbines with compression ratio – turbine inlet temperature curves to 1200 F are actual, above 1200 F they are calculated

Assumptions: turbine efficiency: 0.84; compressor efficiency: 0.83; compressor inlet pressure: 14.7 psi; compressor inlet temperature: 60 F; isentropic compressor exponent: 1.4; turbine inlet temperatures as shown on curves; pressure drop from compressor to turbine: 0

coaches, in the windshield and headlights. When approaching they are at right angles, causing nearly complete blackout. The oncoming vehicle's marker lights will appear normal.

An inside destination sign will advise customers of the next town on the route. Selection of all destination signs will be by push buttons similar to modern car radios.

An automatic fire extinguishing system will be built into the bus. This may be of the CO_2 type and operated by electricity. Probably the fog type will be required where cooling of combustible material is needed: tires for example.

Thus, your author has reported on a conception of "the motor coach of the future," as envisioned by an engineer and not an artist.

No prediction of this nature would be complete without presenting some reasons for the prediction.

The only justification for the existence of a motor coach is its ability to gather in the nickels.

In order to attract customers, and nickels, it is necessary to create a desire on the part of the public to ride in the bus.

This can be accomplished by:

1. Building safer buses, which, we believe, justifies our prediction of nonvolatile fuels, tires without air, better brakes, better methods of suspension, polaroid glass headlight lenses and windshields, power steering, escape sash, and a built-in fire extinguishing system.

2. Building more comfort into the vehicle, which seems to us to justify our prediction of a new improved air conditioning and ventilating system, circulating ice water, wider and longer buses with retiring rooms, inside destination signs, polaroid glass windows, individual radio, pressurized passenger compartments and filtered fresh air, individual draft control, and soundproofed bodies.

3. Building them with more eye appeal, which will require decorations, which must appear inside and out; soft, restful interiors instead of some of the nightmares we are now asked to produce and ride in.

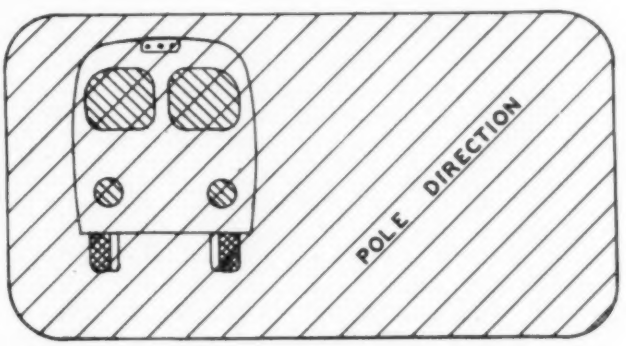
4. Building buses to give the public better service. Under this heading we can list, higher speeds as the result of more power at less cost, the radio telephone enabling a business man to communicate with his office or associates at will, and a radio system permitting him to listen to markets, news, and so on.

5. Making it easier and simpler than driving a private car. This service can be improved upon. How many of you have thought you would like to go downtown, get on a bus, and go to Colorado, for example, on a vacation? Perhaps the only reason you didn't do it was your need of a car after you arrived. Some smart bus company is going to arrange a service which will work something like this: Desiring to make such a trip the potential customer makes his reservation a week in advance and at the same time he applies for the use of a car in Denver. The bus company checks the customer's references and advises him, by phone, his reservation has been accepted and his seats and car have been reserved. He drives his family to the station at the appointed time, parks his car in the bus company's terminal at 50¢ per day. He and his family board the bus, and upon arrival in Denver, he is met by his company car. He signs a receipt, climbs in and drives away. Upon his return he pays a nominal mileage fee on the company car, gets aboard the bus, and is driven home. Upon arrival he recovers his own car, drives home, and everyone has made money, the customer is rested, and everyone is satisfied.

6. Making it cheaper than private automobiles. The city bus is a good example of what we mean by this. Who can afford to drive his own car down town daily when he can ride a bus? But if it is necessary to walk more than a very short distance people will drive, and the number who do drive will increase in direct proportion to the increase in walking distance. So it is with intercity service. It must be handy and frequent to survive. No bus service is good if a man can miss a bus and walk to his destination without another bus passing him on the way.

It is evident, therefore, that all buses will not be 40 ft long, but we believe they will be at least 102 in. wide.

In closing, let me say that it is our belief that this wonderful vehicle we have just described will do all this at 1½¢ per passenger-mile, or very close thereto.



■ Fig. 9 – Direction of pole lines when polaroid glass is used, as seen through the driver's windshield

Factors in A

by W. L. WEEKS
Wright Aeronautical Corp.

AERATION of aircraft-engine oil has spasmodically caused trouble with pressure regulation for several years.

A protracted investigation and test program by Wright Aeronautical Corp., though still unfinished, shows by laboratory, test stand, and flight test, that some facts well known for years have been neglected and aeration of oil therefore invited while deaeration has been definitely restrained.

A review of basic facts and known methods indicates that if they were taken into account in design and service operation, we could go a long way in reducing trouble with oil pressure regulation due to entrained air, according to Mr. Weeks.

The engine itself, obviously, is the main source of entrained air, but the scavenge pumps are not solely responsible, the author reports. Oil fed to them from gear trains contains 6-20% very finely divided air.

Entrained air is, he concludes, inherent to engines with integral reduction drive gears, supercharger drives, and multiple accessory drives.

In facing this fact, Mr. Weeks states further, we cannot continue to handicap deaeration with low operating oil temperatures and localized oil flow through tanks as imposed by the conventional hopper.

With these restraints to deaeration removed, he feels that it will probably be found unnecessary to add accessory operating devices to the powerplant oil system in most cases, although several devices or means either have been or can be developed.

THE AUTHOR: W. L. WEEKS has been in aircraft engine development work since 1917, when he joined the Curtiss-Wright organization in Buffalo. From there, after a period of service in World War I, he went to Wright Aeronautical Corp., where he is now project engineer of the lubrication system section.

AERATION of engine oil in the airplane powerplant has been a recognized source of trouble for the past four or five years. The frequency of trouble cases has increased as engines have increased in size with their oil flow rate stepped up proportionately or more. The study of the problem has been an absorbing one.

Our first experience with air entrainment in the field was not enjoyed, for it came at a time when there was no available means of measuring the air content of oil to an engine pressure pump installed and, therefore, the airplane powerplant people stumbled with us, through the darkness we were in, to two conclusions. They were, that scavenge pumps are the prime source of entrained air, and that the conventional oil tank hopper is not a prime separator. Although the first conclusion was an over-statement and the second a gross understatement of fact, as we now know, they did serve to instigate our serious study of aeration trouble along lines both internal and external of the engine.

This opportunity to discuss a subject which we have actively pursued since June, 1941, comes at a time when we feel very certain of the modifications those four-year-old conclusions must take. I hope to show that the Wright Aeronautical Corp. has maintained the objective approach to the overall problem, recognizing that so many factors were involved that some neglect and shortsightedness were going to be found chargeable to all of the three interested groups, engine designers, airplane powerplant engineers, and operators. We will review as briefly as possible the investigations made in an order of natural relation rather than chronologically because, as many of you know, we learn as we go and often find that today's test results show that such tests should have been made ahead of some we made two years ago.

■ Problem of Oversize Scavenge Pumps

In the engine we must have scavenge pumps which are oversize to a degree that will enable them to clear the sump quickly at a start or following a roll and to keep ahead of engine oil flow even when little atmospheric pressure is on hand to move sump oil into their rotors. It appeared obvious that some way should be found to prevent these pumps from satisfying their excess capacity with crankcase gases at altitudes below their operating ceiling. A review of what had been done and a careful look into what could be done did not bring forth any device suitable as an aircraft-engine component, so we granted this project a long-term status and put other irons in the fire.

Accepting aeration as something we would have to put up with for a while, we took the pressure pump apart and looked at it from several viewpoints. First, was it the best type of pump to use; second, what of its specific performance with oil containing a nominal amount of air and with larger amounts, anticipating cases extreme to various degrees; and third, what should be expected of the pressure regulating (or relief) valve when operating on mixed liquid and gases?

External of the engine the lubricating oil system consists of a group of items, the design and operation of which are a matter of divided or correlated responsibilities. Never-

[This paper was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Calif., Oct. 4, 1945.]

in AERATION and DEAERATION of AIRCRAFT-ENGINE OIL

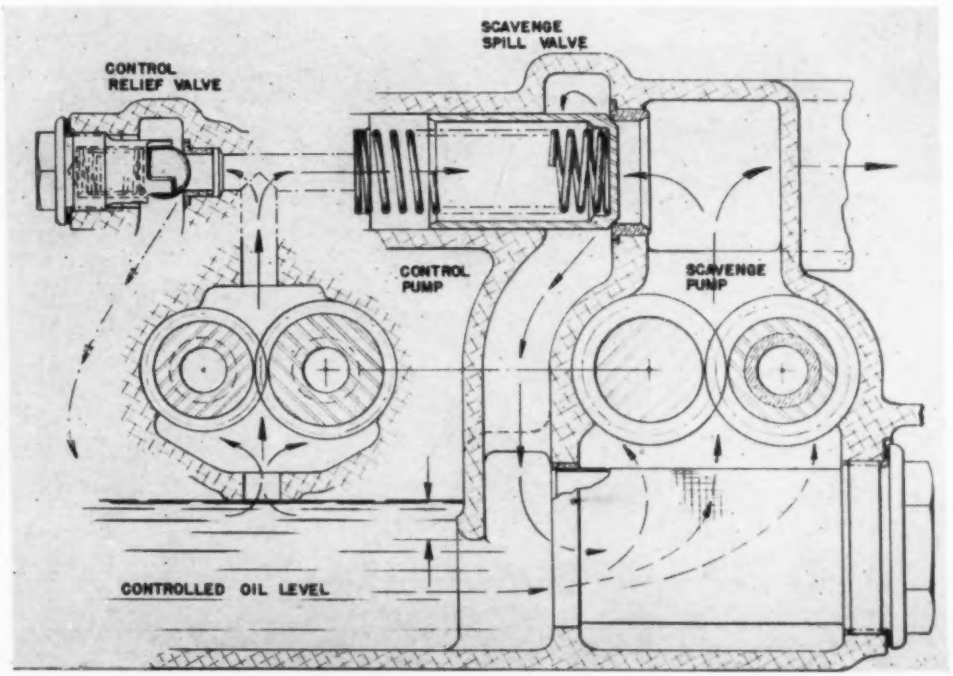
theless, since the engine is dependent to some extent upon the outward part of the system for the return of oil in a usable condition, it was felt that the engine company should know about its operating factors and characteristics. Studies were, therefore, made of oil tank designs of both military basic types, their air-separating characteristics compared and operating conditions and factors investigated. Auxiliary air-separating devices were investigated, designed and tested, or procured and tested as stop-gaps or crutches which might be used if engine and airplane normal components were not found susceptible of the required degree of improvement.

Although there are some items of development and investigation yet to finish, we believe it is clear right now that several important factors in the aeration and deaeration problems have been overlooked and too long neglected, although some of them are well known. Tests show that a very large degree of improvement can be made if a small stock of basic facts is capitalized.

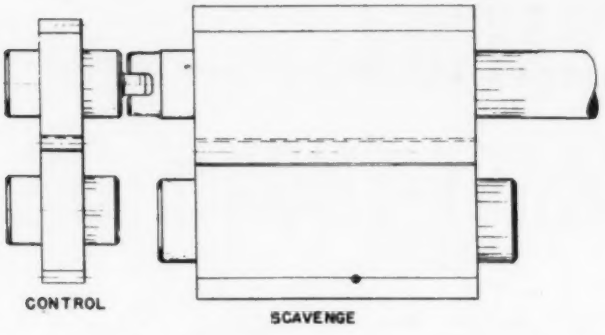
This paper treats the factors in aeration and deaeration, about which something can and should be done, as having trends rather than showing all data quantitatively for specific cases. The trends are obvious and undeniable, whereas specific data from service installations are often involved with conflicting or opposing effects. In the flight test work done in our investigations we were fortunate in having a free hand in setting up test conditions and more fortunate in having an airplane whose standard powerplant had a really good case of aeritis. Oil pressure loss and fluctuation would occur almost before the ink was dry on the flight schedule. This gave an opportunity to work down a condition, at its worst, of about 24% entrained air as measured by a Wright-Gulf indicator.

■ Fundamental Concepts

Before presenting facts and figures it may be well to state some concepts which we settled upon very early in our work, and which are, we believe, also held by NACA.



■ Fig. 1 – Hydraulic type of scavenge control



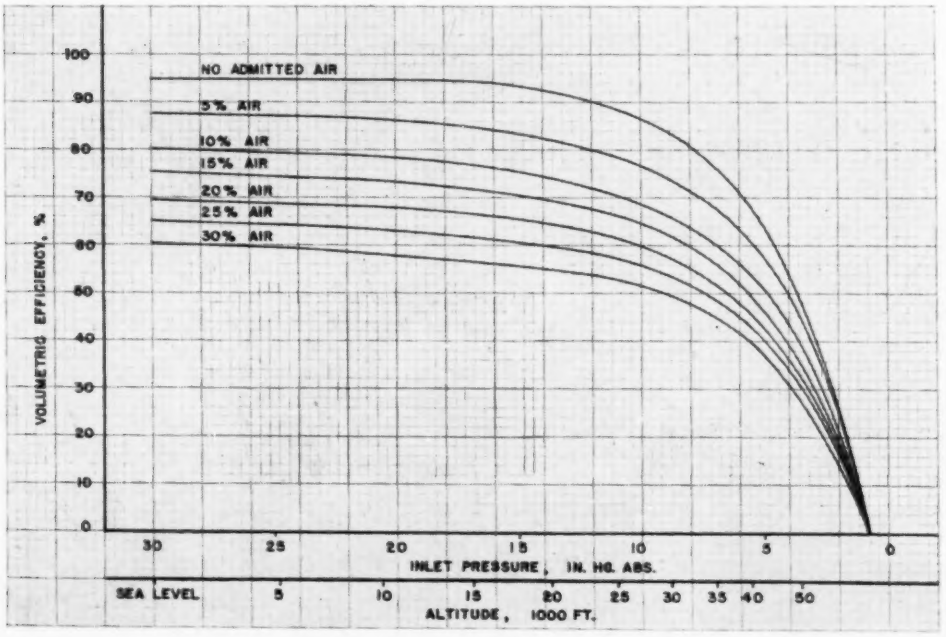
■ Fig. 2 – Relative size of rotors for hydraulic scavenge control unit

Gulf Research & Development Co. and most others who have taken active interest in the oil aeration problem. Entrained air is accepted as free air masses of any size moving with the oil stream in tanks, engine sumps, lines, or passages. The same masses, if small enough to be held by enveloping oil film at the surface of oil in a tank or sump, would be termed foam or froth. Entrained air traveling in the oil stream in lines of some substantial length is known to rise by flotation and, subject to oil flow velocity, sooner or later striate the fluid stream and move at a velocity of its own along the top of the line, either as a stream of bubbles or as air slugs. Whatever the structure resulting, the air content of the flowing mixture is still called entrained air, on the score that it is air carried in or with the stream on its way to be a curse or to be separated. We believe the word foam is not correctly used as naming the air in a flowing line. In isolated cases where entrained air is released in a tank at such a high rate that, in a state of foam or froth, it is carried by turbulence in the tank to the tank outlet, it becomes entrained air in the exit line. In expressing the per cent entrained air it is taken as per cent by volume of the *mixture*, in other words, air volume divided by oil plus air total volume.

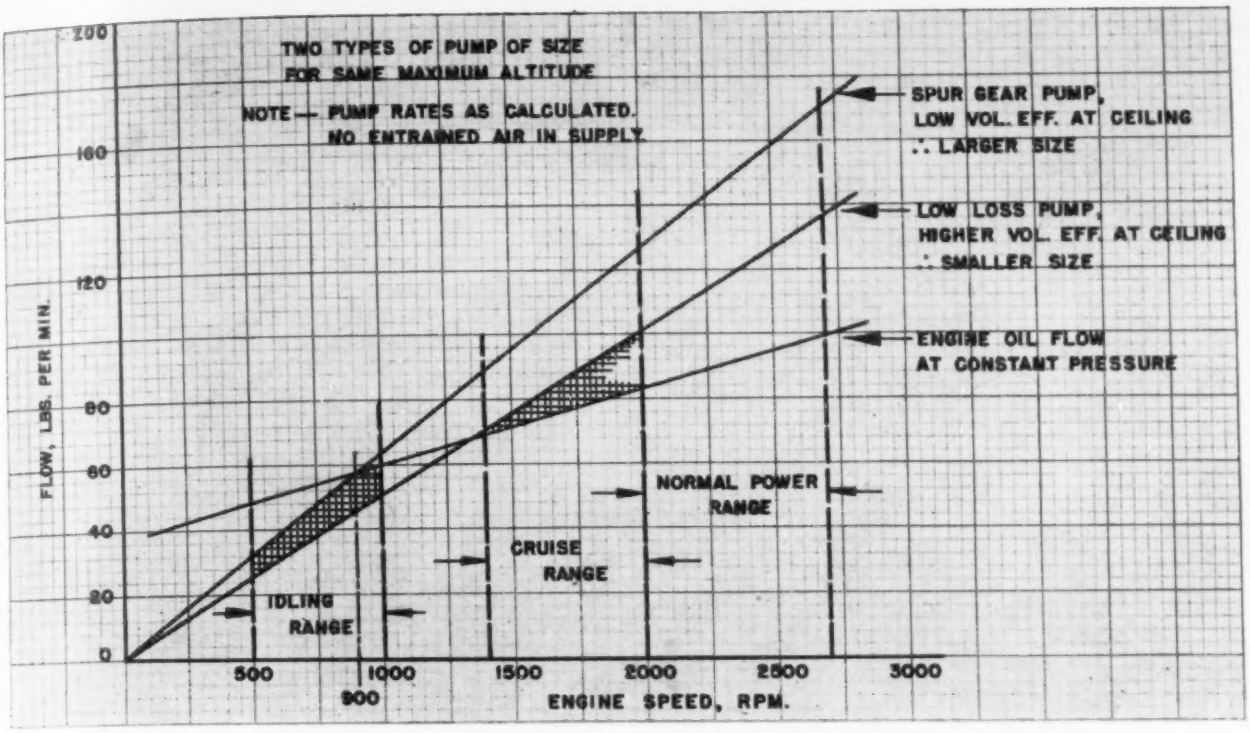
■ Controlled Spill Valve

Taking, now, the work incited by our early and quite nasty experience with entrained air, we have to report that one fairly acceptable means has been developed to keep scavenge pumps from "inhaling" crankcase gases. It is basically one of the potential dozen or so ways to keep that pump's inlet submerged in sump oil. Fig. 1 shows the device in principle and general proportions. The principle is simply that of controlling the opening of a spill valve in the outlet passage of the scavenge pump so that oil is moved from engine to cooler at exactly the rate of engine oil flow into the sump.

This result is accomplished in the present device by hydraulic pump and plunger action, the spill valve being actuated as the plunger. The inlet of the hydraulic, or control, pump is located a fraction of an inch higher than the inlet of the scavenge pump. When sump oil level is lowered to the control pump inlet, air is drawn in and hydraulic pressure on the spill valve drops, allowing spill to occur and thus rebuild the sump level. This results in the sump level being maintained slightly over the scavenge pump inlet so that pump runs constantly submerged but discharging to the oil cooler at exactly engine flow rate. In the hydraulic line a relief valve is provided to limit pressure to that required to counter, with a small margin, the maximum scavenge back pressure allowed by specification or imposed by winter operation. It is, therefore, set to about 42 psi or higher if cold-starting back pressure is controlled by a surge valve to protect the oil cooler. Upon engine starting, both pumps submerged, this valve passes the full flow of the control pump, there being no other outlet except the plunger clearance of the spill valve. This clearance serves to relieve air pressure when the control pump is functioning normally and discharging part oil and part air. Fig. 2 illustrates the small size of the control pump in relation to the scavenge pump.



■ Fig. 3 – Effect of entrained air on pump volumetric efficiency for a typical spur-gear pump



■ Fig. 4 – Pump output versus engine flow required

The device is positive in action and will not surge or hunt if the control pump inlet is made small and the control pressure relief valve and its passages reasonably generous. Kept submerged, the scavenge pump adds no air to scavenge outlet oil but delivers sump oil as such, with whatever air content that oil contains.

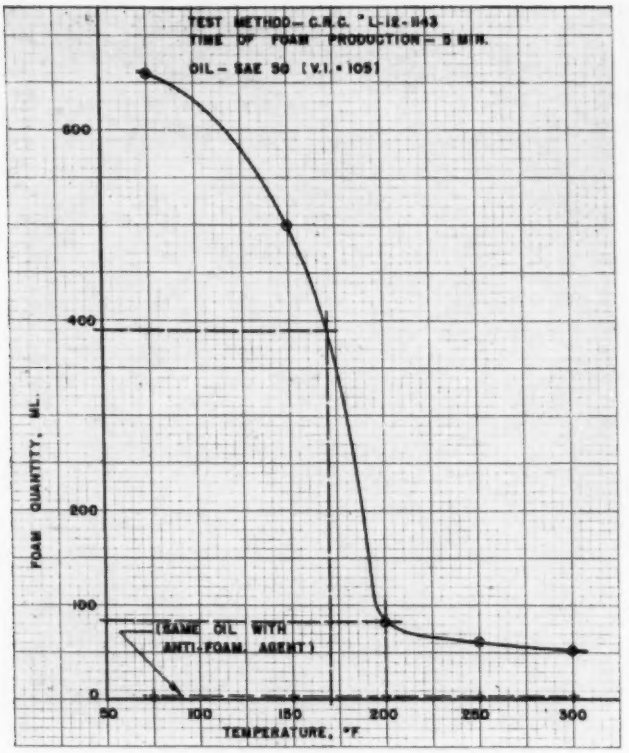
Except in rare cases the device cannot be very well applied to engines unless original provisions are made for it. However, when included in a new design it takes very little space, as we found when we applied it experimentally to two late engine models. Another type of control which consists of only two moving parts is currently under development.

■ Pressure Pumps

Parallel with work on the scavenge system the pressure pump has had a pretty thorough going-over. We simultaneously examined the conventional spur-gear type for inherent characteristics and means for improvement and made tests and/or investigations of other types. The first job was to compare one of our own pressure pumps with a good commercial pump made by a concern long in the business and whose work is of high grade. The comparison was quite close. Both pumps had some unnecessary flow restriction in their inlets, sharp turns in our case, and too small an inlet in the case of the commercial pump. In our case the resistance mattered and we have since designed for the maximum freedom of inlet passage. In the commercial pump the loss was of no importance because the pump was not expected to be used at altitudes and if some service at depressed inlet was intended, the manufacturer would simply recommend a larger model, and the customer would never blink at double the weight.

The two pumps reacted very much the same to air entrained in their supply. A good deal has been written on the effect of air entrainment and some very good papers and expositions on the basic characteristics of gear pumps also have been presented in the past four years by J. Dolza of Allison Division, R. J. S. Pigott of Gulf Research & Development Co., and P. H. Schweitzer of Pennsylvania State College. Fig. 3 is, therefore, shown merely as a refresher. With the conviction that at least some entrained air is encountered in service, it is obvious from the graph that any other handicaps, such as excessive rotor tip speed and inlet lines, restrictive because of size or sharp turns, should be avoided.

Development tests of spur-gear pumps have proved to many that, reasonably well built, the type is not unsuited to the job of supplying oil pressure in aircraft engines. Our investigation has, nevertheless, covered by test several other potentially good types. In some, centrifugal force is put to work to aid rotor filling instead of being allowed to oppose it. In others centrifugal force is merely ineffective either way, but tooth entry loss is inherently very low. Some advantage can be taken of types having extremely small inlet losses on the point of space occupied. The gain is not, however, of the order represented by their ability to maintain good volumetric efficiency away out to 3 or 4 in. Hg absolute, as some of them do. It works out that if one of these low entry loss types is designed as small for some high altitude as its characteristics indicate, it is quite likely to be too small to do a good job at low engine speed. This is shown graphically by Fig. 4. Here we have an actual case in which two pumps are selected for size to meet an extreme altitude requirement. One is the conventional spur-gear type and the other is one of the two best types having low entry loss. The output lines are drawn to



■ Fig. 5 — Temperature effect on foaming

calculated pump flow rate, 10% entrained air taken into account since they would be about equally handicapped by whatever amount is encountered.

As chosen to take full advantage of its higher efficiency at low inlet absolute pressures, the low loss pump is naturally the smaller. In the engine speed range where normal power is used it is seen that both pumps have flow rates in excess of the demand and normal oil pressure will be maintained by the pressure relief valve. In the range of cruise engine speeds, however, the low loss pump is found to have a marginal output around 1500 engine rpm, and with the nominal amount of entrained air assumed oil pressure would be subnormal even at moderate altitude and would worry the pilot or flight engineer.

Since the engine will take a definite oil flow at constant pressure even when standing, no engine-driven pump can possibly maintain that pressure in the low idling range. In the present example the spur-gear pump could maintain constant oil pressure down to 900 engine rpm. However, the low loss pump, because of its smaller capacity, could maintain the pressure only down to 1350. In service some order of low oil pressure is expected but frequently questioned as to being acceptable.

■ Compromise in Pump Size

From the analysis of several cases, such as Fig. 4 illustrates, it is plain that in taking advantage of the efficiency of the low loss pump a compromise has to be made in pump size to obtain a conservative supply of oil at low speeds. With this compromise made, some of these pumps may appear in new engine designs, incorporated as a factor in some economy of space and weight. On the whole, our investigation of pump types, although not quite completed, indicates that the spur-gear pump is not a bad choice that has persisted in the aircraft-engine industry.

■ Pressure Relief Valve

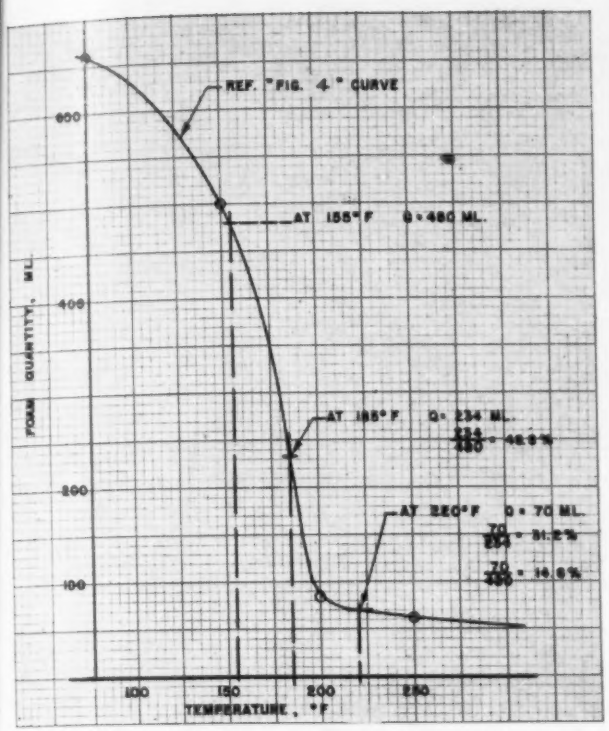
Sharing the responsibility for maintenance of engine oil pressure with the oil pump, the pressure relief valve was a natural subject included in studies of pump behavior. A simple enough device to be quite trouble-free, this unit of the engine oil pressure system has played the part of "whipping boy" in a great many cases of field trouble with oil pressure in the past and present. It is quite pertinent that all field trouble with relief valves has consistently been found concentrated in operating conditions of either extremely dirty oil or entrained air of a quantity order unacceptable otherwise to engine operation.

The "whipping boy" role of the pressure relief valve is very old, as illustrated by a story our Roland Chilton tells. It has to do with a workhorse airplane used by our flight test group about 15 years ago and which could be flown only about 20 min at a time before oil pressure fell off. It was quite obvious that the relief valve was the cause, for a routine procedure proved it. The routine ran thus: In case of loss of oil pressure in flight, land, remove and clean the relief valve, reinstall and adjust and resume flight. Naturally, the test pilot lit a cigarette while the mechanic cleaned the valve and put it back.

The persistence of the trouble caused Mr. Chilton to go out one day and ask for a look into the oil tank the moment the cap could be removed when the pilot came in for the routine cleaning job. Having seen the condition of the oil in the tank, he suggested that the routine could just as well as not be simplified, thus: "In case of loss of oil pressure in flight, land at the nearest field, get out of the plane and smoke a cigarette. After completing this operation, again enter the airplane and start the engine. It will be found that the trouble has been fixed." That was found to be correct. The look into the tank had shown that the scavenge return produced a perfect cascade and 20 min of running had charged the entire tank content with air to the point of almost overflowing when the tank cap was removed. Prediction of the behavior of a relief valve working on a mixture of oil and air is practically impossible. When we consider that it constitutes a spring-loaded variable orifice throttling a mixture of two fluids, one of which has a density, roughly seven hundred times greater than the other, this is not surprising. Also, it is understandable that the simple device is confused as to what to do when air reaches it in the form of elongated slugs, as is common when the total air entrained is 15% or more and the supply line run is four or five feet long.

■ Test Stand Experience

Our experience with the effect of entrained air on pressure regulation has not been solely in airplane installations. Test stand oil systems have been found to be poor in air-separating function in several instances. In one of these we encountered a phenomenon which is often difficult to recognize and undoubtedly is involved in some cases of oil pressure trouble in flight installations. In a group of test stands having oil tanks too small for the oil flow rate of the particular engines tested on them, the supply lines fed about 16% air. Suddenly, after satisfactory use of the stands for several months, an epidemic of oil pressure drop was experienced. It occurred only at a very narrow engine



■ Fig. 6 – Temperature effect on foaming – temperatures in range of service conditions in airplane oil tanks

speed range, say, two or three hundred rpm. After a lot of trouble shooting and record checking it was found that a certain minor pattern change had been made in the oil pump. The change had eliminated a blind branch in the cored discharge passage of the pressure pump. Since the date of release of the first of the new castings coincided with the start of the trouble, one of the old style castings was tried on an engine. The pressure fluctuation disappeared. Experiments with blind drilled pockets in the new castings were then made on the assumption that the compressibility of a 16% air volume in the oil invited column vibration in the pressure oil passages. A single, short-drilled hole was found to provide sufficient damping. Changes in relief valve mass and spring rate had been found to have little effect. This circumstance is cited to show that there is more than one incentive to better control of air quantity. Compressibility is an unwanted factor in any liquid pressure system.

We would hesitate to say that it is impossible to design a pressure regulating valve which would be indifferent to wide variations in density of the fluid it throttles and insensitive to resonant vibrations set up in the branch lines to it, due to the compressibility of the fluid in the system. Our experience to date indicates, however, that the degree of air entrainment which would justify such a valve is one which cannot be tolerated in its effect upon volumetric efficiency of the pressure pump.

It may be reasoned quite logically, therefore, that the conventional type of relief valve is a good pressure regulator when oil condition as to entrained air is good, and when there is too much air for normal pressure system operation it serves as an indicator of that condition. Our efforts in relief valve development are focused on alleviation of abrasion in service with the worst conditions of oil as to entrained solids – the dirtiest oil encountered.

■ Oil Tanks

In order to learn what degree of air separation should be expected of airplane oil tanks of accepted design we selected, over a period of two years, four tanks which we considered typical. The tanks were set up with the actual airplane lines, oil coolers, and fittings, and run with the engine oil pumps, pressure and scavenge, used in the installation. The scavenge pump took oil from a tank representing the engine sump, that oil being continuously charged with 10 or 12% air as sump oil was considered to be charged. The test results were viewed in the perspective of laboratory tests in oil foaming made by others as well as ourselves.

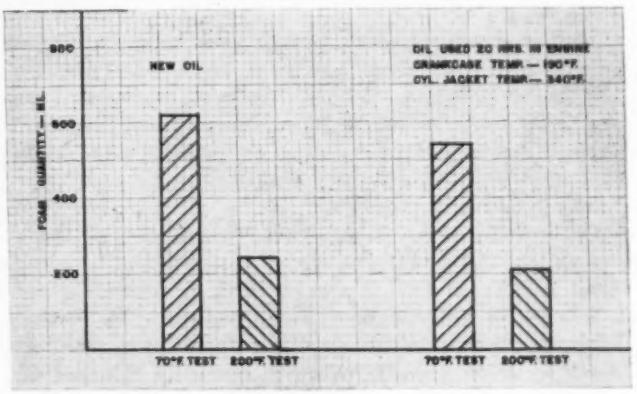
These background tests are far from being merely academic so we wish to review them on the basis that factors which influence the production of foam in a simple process and the stability of foam when produced are clearly demonstrated.

■ Tests by Others

In a paper by Ambrose and Trautman,[1] some tests of temperature effect and the effect of engine service time on foaming and foam collapse are described. In these tests use was made of the simple CRC apparatus reported by the Group on Foaming under Designation L-12-1143 in November, 1943. In it a controlled supply of air is blown through a porous stone diffuser at the bottom of a 1000-ml graduate. Oil volume is initially 200 ml and foam volume is read as produced in 5 min. Foam collapse is evaluated from the mixture volume read after a 10-min lapse after air shut-off.

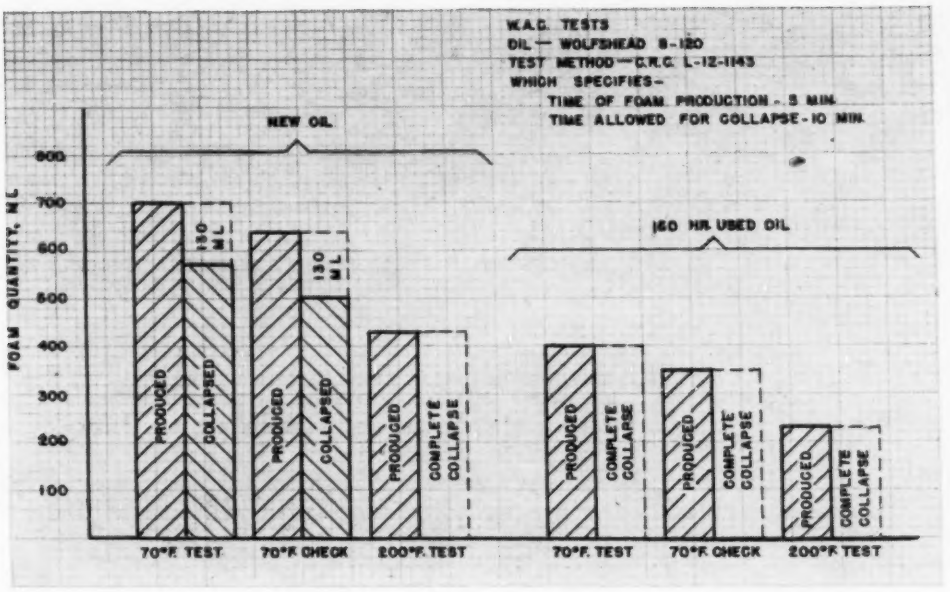
The authors' Fig. 4 (right), reproduced here as Fig. 5, shows the effect of temperature in the case of an oil not greatly different from a commonly used aircraft-engine oil in viscosity index and like it in not having detergents or other additives. It will be noted that only one-quarter as much foam was produced at 200 F as at 170 F. The subject of antifoam agents is treated briefly elsewhere in this paper but it is interesting to see shown the effect of one of those agents as practically stopping foam production in the reference test.

Fig. 6 views reference Fig. 4 in the light of oil tempera-



■ Fig. 7 – Temperature effect on foaming – CRC L-12-1143 test method; time of foam production: 5 min

[1] See SAE Transactions, Vol. 53, July, 1945, pp. 373-376: "Engine Oil Foaming," by H. A. Ambrose and C. E. Trautman.



■ Fig. 8 – Temperature effect on foaming and collapse of foam

tures at the tank in service airplanes. The 155 F represents the case of an airplane model active in large numbers this year, while 185 F is the temperature at which its engines are designed to operate. At the latter temperature engine oil flow rate and heat rejection are normal. Please note the wide difference in foamability at 155 F and at 185. The 220 F point is chosen for discussion only because beyond it the instability of foam is not greatly increased. It is a temperature, however, 20 to 50 F below that at which oil tanks would function as air separators if we put the oil coolers downstream of them.

Other tests reported in the reference paper indicated the effect of temperature and that of engine service. Our Fig. 7 is plotted from the paper's Table 3, and while the low temperatures are extreme in terms of aircraft operation, it is noteworthy that the oil's foaming tendency is reduced with service.

The paper also discusses foaming as it is encountered in gearbox units, such as automotive transmissions, including differentials. It states that in some instances foaming has been so severe that sufficient lubricant was lost by being forced out of the housing to make operation dangerous. No scavenge pumps are involved in such cases. The foam is produced simply by the "milling" action of the gears on the oil in the presence of air.

We made some tests of the tendency of engine service to "wear down" the foamability of oil, the results of which are shown in Fig. 8. If they were to be done again we would include 150 and 185 F and deviate from the CRC method to time the collapse of foam produced rather than give it an arbitrary 10-min period. That much time is not available in any airplane tank.

A few years ago the Royal Aircraft Establishment[2] did a "good bit of looking into" the business of oil foaming, froth stability, and those characteristics as determined for a number of different oils. From this work we present Fig. 9 as a partial replot of foam collapse times of an oil lower in viscosity index than our commonly used 120-sec oil but covering a temperature span that is pertinent. The apparatus used is similar to the CRC one, but foam collapse is timed. You will note that the initial volumes of foam were not identical, being 215 and 235 cc, but we transpose the larger for a direct comparison, admitting a small error may be introduced. Conservatively it can be said, then, that the collapse of an 88-cc volume takes place at 229 F in 12 to 15% of the time required at 131 F.

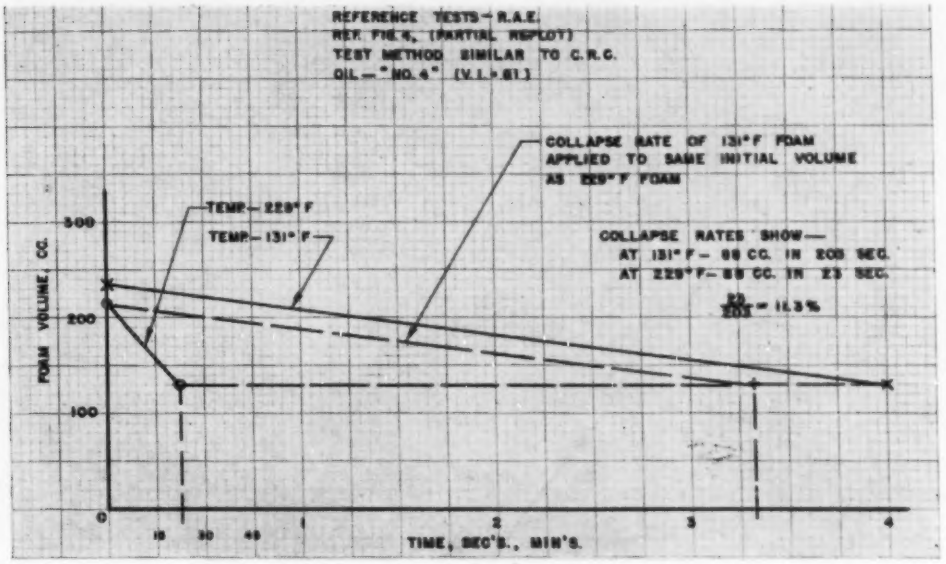
With the relation of temperature to the foamability of oil and to the stability of foam as our principal leads, we made the forementioned tests of some production oil tanks. Two were of the hopper type and two of the type that makes use of a quick warm-up compartment which is supposed to be automatically valved out of the circulated oil route when some temperature called normal is reached. These tanks were tested with that latter condition assumed, full flow to the main compartment of the tank. Fig. 10 identifies these two types as *A* and *B* for reference to curves which follow. For the present discussion the specific design of warm-up compartments in the *A* tanks can be ignored. In the *B* tanks the hopper inlet fittings were much alike in oil spreading action, so we call them comparable on that point.

■ Problems with Conventional Hoppers

Two of our most troublesome service cases of oil pressure drop or fluctuation, found to be definitely a matter of high air content in the oil supply, involved conventional hoppers. The worst case in any of our flight test airplanes was one having a nearly conventional hopper. The case on which the Wright-Gulf air indicator was first used showed the highest air entrainment we have ever encountered and that airplane had a conventional hopper type of tank. Although the engine was not ours we welcomed the opportunity of experience with the indicator.

It was natural that we should want to know something about the hopper. Fig. 11 is the plot of oil temperature effect on air separation in a *B* tank. We can state that the trend is representative of all tanks of this type. Extrapolation in this particular case is seen to indicate a reduction from 13 to 6% air as oil temperature is changed from 125 to 185 F. Although we know that bubbles burst most

[2] See "Aeration and Frothing of Lubricants." Report of the Royal Aircraft Establishment, Farnborough, England.

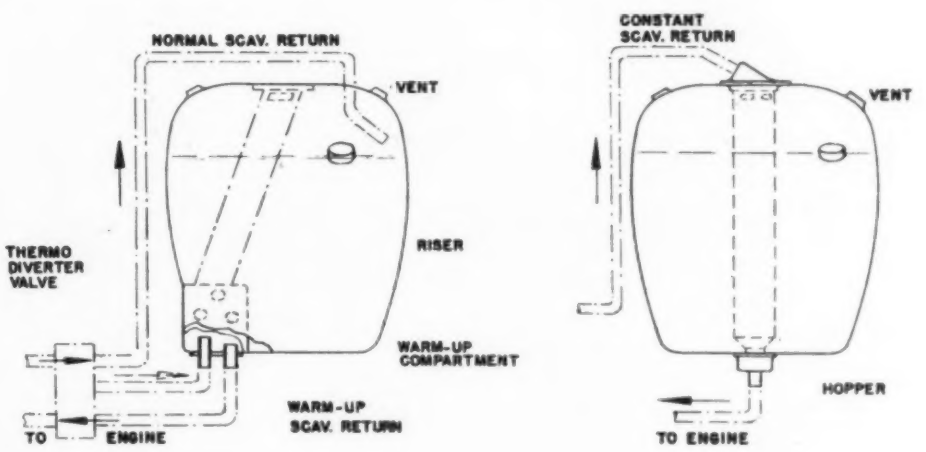


▪ Fig. 9 – Temperature effect on time of foam collapse

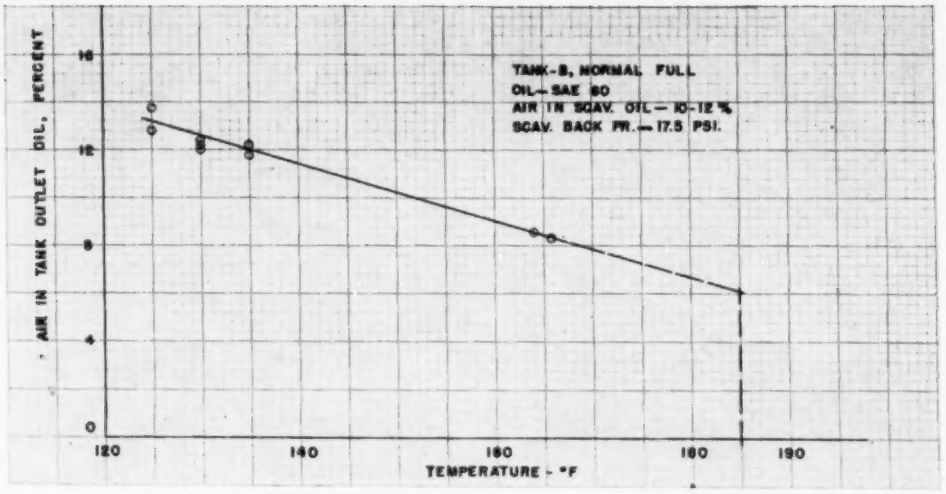
readily when their skin is weak from lowered viscosity, we plotted Fig. 12, which refers the gain of 7% separated air to a change in viscosity from 740 to 187 SUS.

In the *A* or open type of tank, our best references were obtained from flight tests and will be shown later. In the light of the laboratory tests of foam stability it seemed logical that anything affecting time available for flotation and surface release of entrained bubbles should be appraised. The rate at which oil passes through a tank is, therefore, of interest. In an *A* tank, if the inlet and outlet are remote, as they surely should be, the mean rate of through flow is low. In a hopper it is quite otherwise. We will use an abstract expression for the amount of oil in tank circulation in relation to rate of flow out of the tank by dividing the former by the latter and calling it factor *F*. Using this terminology a 27-gal capacity *A* tank serving an engine with 80 lb per min would rate a number above unity and a *B* tank hopper serving the same engine would rate a small decimal. Fig. 13 shows air separation in an *A* tank with oil temperature 185 F plotted against the *F* factor as it varies with tank oil level. The trend line is what could be expected, larger volume of oil, slower traverse, therefore, more time for flotation and release of air at the oil surface.

Fig. 14 is a similar plot for a *B* tank in which, you will note, *F* is 0.075 at full tank condition as contrasted with 1.33 in the *A* tank case. The 165 F temperature is an average service condition for the installation using this *B* tank. It is plain that the height of the oil column moving through the hopper has little effect on air separation. This is in spite of the fact that more free sidewall is available for the scavenge oil stream to spread upon. So we concluded that most of the air that is not separated is too finely divided to agglomerate and burst on the free sidewall and is carried on down. Aside from some jet energy that the scavenge mixture has when it enters the oil level, more at high tank levels than at low ones, there is the nominal velocity of vertical flow of the oil body in the



▪ Fig. 10 – Two types of oil tanks



■ Fig. 11 – Temperature effect on separation of air in hopper type of tank

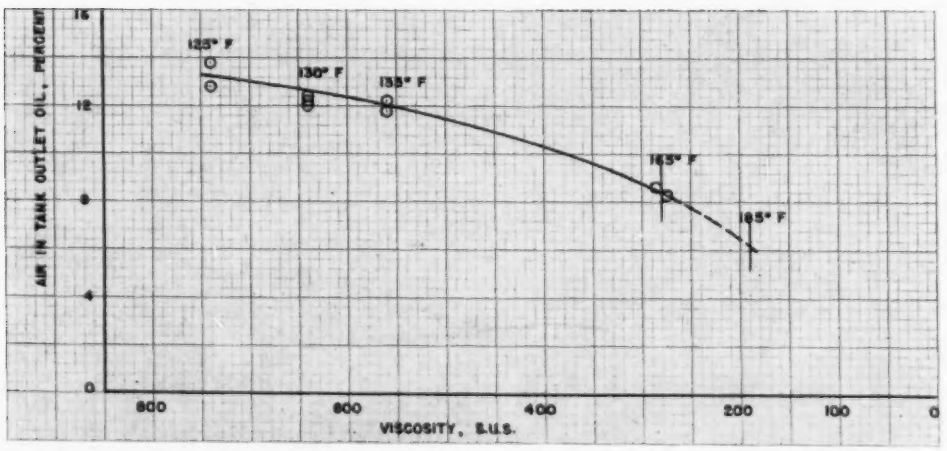
hopper. The two *B* tanks tested were calculated to have velocities of 2.54 and 4.0 in. per sec. The lowest figure is a rate considerably higher than the natural rising rate of bubbles as fine as constitute a good portion of the total air in the tank outlet with oil even at 185 F, and with lower temperature the case is more acute. This is a repetition of the statement we have made on other occasions in only slightly different form, namely, that so far as separation of finely divided air is concerned the conventional oil tank hopper is simply an enlarged section of a line returning scavenge oil to the pressure pump.

By use of parenthesis in Fig. 14 we have shown what happened in air separation when the scavenge mixture was routed to the top of the same *B* tank and the maximum advantage thus taken of volume as giving time for bubbles to rise and of surface area for dissipation. The *F* factor, naturally, does not apply to the dash line.

■ Scavenge Back Pressure

Another factor in aeration, and subsequently in deaeration, was found, both in the tank tests and by full-scale test stand experience, to be scavenge discharge pressure, usually spoken of as scavenge back pressure. It may seem naive, in view of the trouble experienced with scavenge line connections in service, to point out any further handicaps of high pressures. The specification limit of 40 psi has, we all know, an older reason than safety of lines, namely, assurance against stalling scavenge pump flow. This reason no longer holds, for we know how to make scavenge pumps insensitive to the order of their discharge pressure. Nevertheless, against the possibility of a flexible and superpressure tubing connection and oil coolers which will not rupture coming into use, we stand up to say that as scavenge back pressure is increased, entrained air separates less readily. Further, it can be stated that if restriction to scavenge oil flow induces turbulence, air separation becomes still more difficult. This is borne out in engine tests where two different oil cooler designs imposing the same back pressure on the engine scavenge pumps resulted in different percentages of entrained air in the oil tank outlet.

We saw that the nature of restriction as well as the order of it has a bearing on eventual separation when we set up an *A* tank and ran a few tests before we had received the oil cooler, lines, and fittings. For these preliminary runs



■ Fig. 12 – Relation of oil viscosity to separation of air in a hopper type of tank

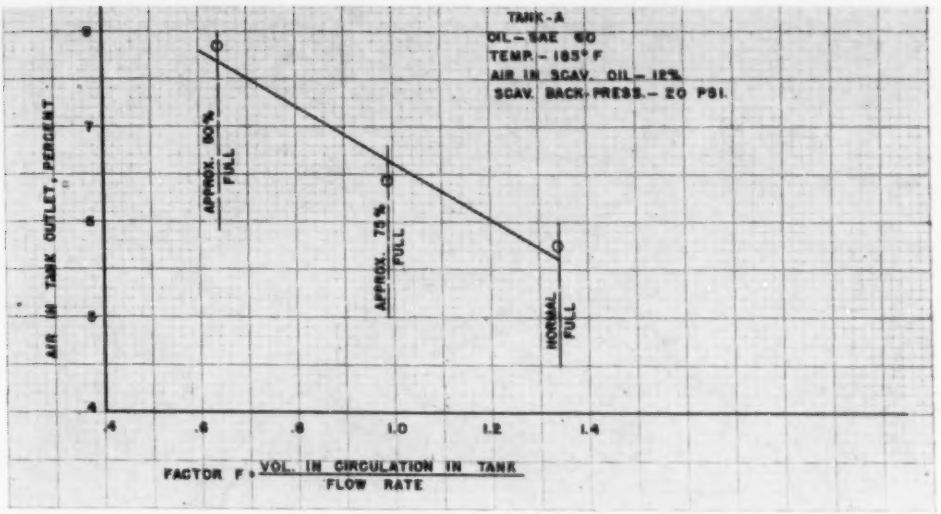


Fig. 13 – Effect of volume of oil in circulation in tank on separation of air

we used tubing of the size and length used in the installation and to impose various back pressures we clamped the hose connection nearest the tank to flatten it. When the actual complete assembly was tested the results did not check very well with the previous ones. Fig. 15 illustrates this with an arbitrarily three-quarters-full tank and oil at the temperature it is intended to be. It is interesting to see that something of the nature or order of turbulence at given pressures causes the curves to cross. It is more interesting to see that, with the actual oil system any superficial throttling causes a rise of entrained air values on an almost 45-deg slope. Back pressure at normal engine conditions including 185 F oil-in was about 16 psi. It is obvious from the solid line curve that any unnecessary restriction, such as a sharp 90-deg turn in the line, would have been detrimental.

Fig. 16 reverts to the subject of the effect of oil volume in circulation or tank level effect on separation in the same *A* type of tank and points to the less complete separation of air at the tank at high back pressures and small oil volumes.

Oil Supply Temperature

Before discussing the flight tests in which we applied some of the potent factors in deaeration which were investigated on the bench we would add another comment on the point of oil supply temperature. The matter of entrained air being expanded, at the approach to the pressure pump, by pressure drop in the inlet line has been stressed a great deal and at present these lines are being made of generous size. However, viscosity drag can still be an aggravation to any entrained air condition. In the temperature range from say, 165 F downward, even a nominally good sized inlet line will have considerable flow resistance. Fig. 17 illustrates this in an assorted collection of inlet suction readings covering some minor plumbing changes in a flight test airplane. The trend shown is clearly one of increased superficial altitude imposed on the pressure pump by decreased oil temperature and hence increased viscosity. Then a difference of 10 F, that is, 160 to 150, imposes a handicap of 2.1 in. Hg on the pump, equivalent to 2000 ft at 25,000, as well as causing a normal 10%

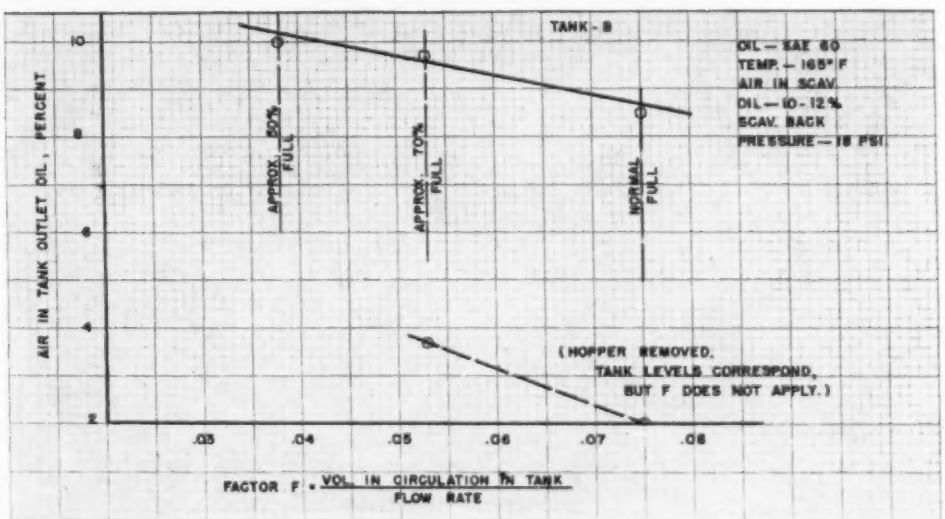
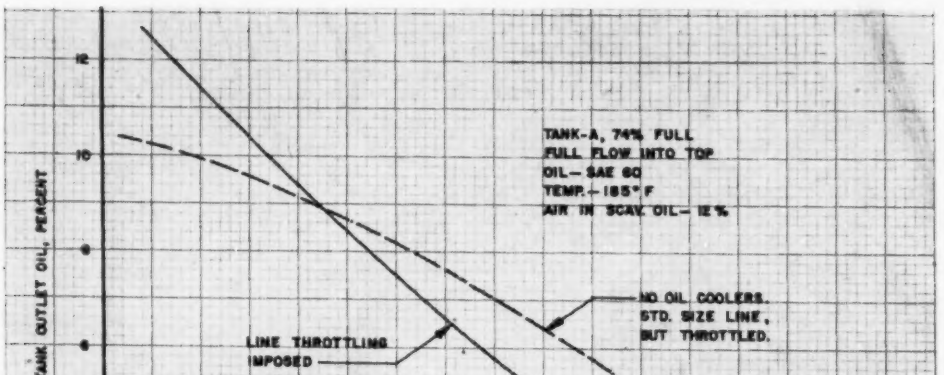


Fig. 14 – Effect of volume of oil in circulation in tank on separation of air



■ Fig. 15 – Effect of flow resistance in scavenge discharge on separation of air at tank

volume of entrained air in the line at tank outlet to be expanded to 12.3% at the pump inlet at that altitude.

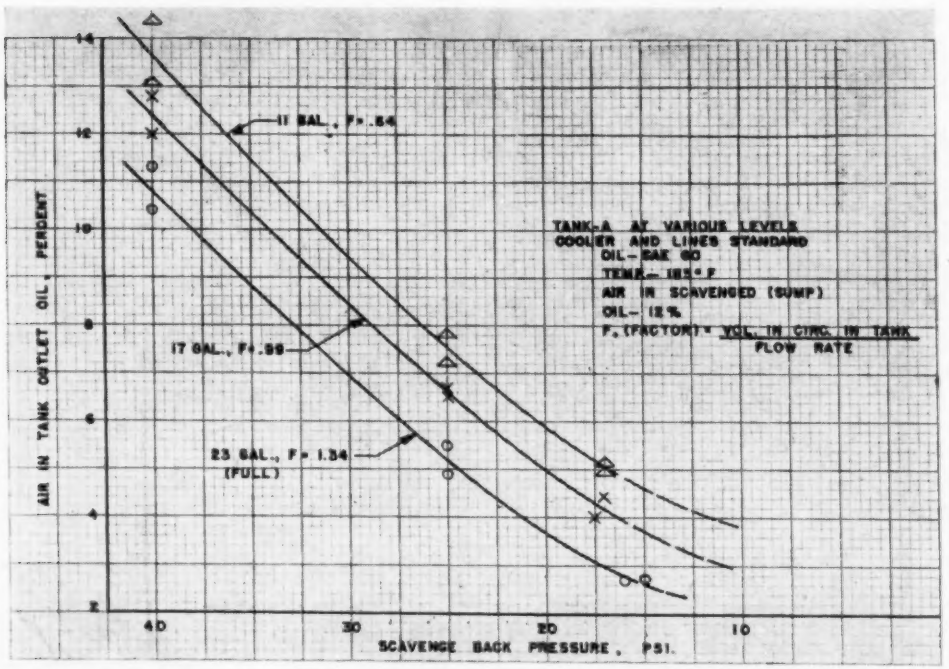
■ Laboratory and Bench Test Results

The laboratory and bench test results viewed in the light of service troubles due to entrained air instigated work along six lines in the past year and a half as follows:

1. Investigation of engine sump oil condition as to entrained air content and fineness of division.
2. Design and test of two types of power-driven centrifugal separators and several kinetic and static types.
3. Cooperative design of an oil boost pump for tank mounting.
4. A survey of oil supply temperatures in field operations to determine the scope of the low-temperature practice, and reasons for it.
5. Investigation tests of long-flow-route oil tanks.
6. Evaluation of the hot tank oil system by flight test.

The outcome or present status of these activities, briefly stated, will qualify our convictions as to what can and should be done about aeration and deaeration.

The responsibility of scavenge pumps, in their excess capacity, for the unsatisfactory separation of entrained air at the tank is not established. We have sampled oil continuously from the bottom of engine sumps half an inch below the surface of oil flowing *into* the scavenge


11 GAL., F=.54
17 GAL., F=.99
23 GAL., F=1.34
(FULL)
TANK-A AT VARIOUS LEVELS
COOLER AND LINES STANDARD
OIL – SAE 60
TEMP. – 185° F
AIR IN SCAVENGED (SUMP)
OIL – 12 %
F. (FACTOR) = VOL. IN CIRC. IN TANK / FLOW RATE
AIR IN TANK OUTLET OIL, PERCENT
SCAVENGE BACK PRESSURE, PSI.

■ Fig. 16 – Effect of flow resistance in scavenge discharge on separation of air at tank

pumps on several of each model of engine. Varying with engine model and proximity of the sump to reduction gears and accessory drive gear sets, the air content of sump mixture, before being scavenged, is known to be anywhere between 6 and 20%. Please bear in mind that we are not talking about oil standing in a well. At anything like a static condition it probably would not retain more than 6 to 9%, being too hot and thin. But this mixture is in flow and has not time to permit flotation of much of its air before being scavenged.

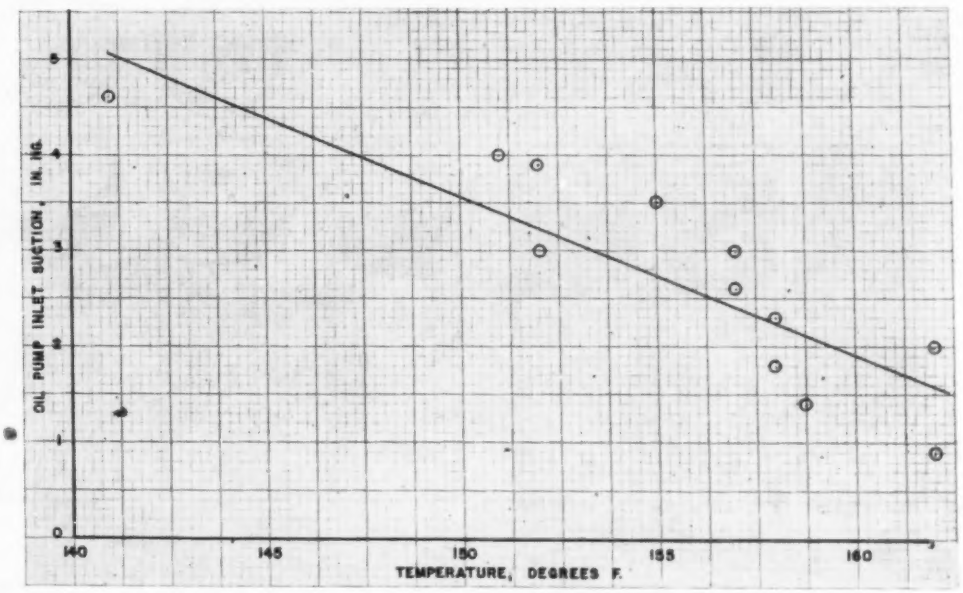
It is not surprising to find this condition. High-power reduction gears, two-speed superchargers, and the expanded list of engine-driven accessories involve the use of a lot of cog wheels. A late engine model has a total of 110 gears of which various groups engage their teeth at velocities of 5500, 8200, and 12,200 fpm. The tip speed of oil pump gears is seldom more than 1300 fpm, so, even though those pumps handle, positively, large air volumes it is likely that the air is not torn into such fine bits as that in the spume from gears meshing at 4-9 times that speed. The most finely divided air is certainly the most reluctant to separate, so it is reasonable to assume that it is high-speed gear trains which produce the emulsion that persists at the outlets of oil tanks. It would be interesting to prove this assumption conclusively but it is not really necessary. Since gears are likely to remain a popular means of transmitting rotative motion there is not much we can do except try to be smart in handling the emulsion. The advent of other rotative prime movers for aircraft than the reciprocating engine will probably not alter the problem greatly. They also will have reduction gears and accessory drive gear sets with high meshing speeds and will turn out scavenge oil in emulsion form. The nature of the emulsion as to fineness of division of its air content is a factor of basic importance and difficult to evaluate. When engine oil has had a few hours of service it holds enough carbon to make the accurate reading of settling levels practically impossible. After some experimentation we accepted, temporarily, a rather unscientific method of gaging the final reluctancy of settling. It is somewhat like using for a scale a stick with rags tied on it at increments. We will soon do better. In watching a sample of scavenge oil taken by overflowing a graduate held under the return line at the tank on a test stand, we struck on a rough criterion. As coarse foam collects and disperses on the surface and the fine bubbles continue to rise but at a progressively slowed rate, the total foam mass shrinks to expose the oil surface. A tiny spot grows larger, and almost invisible small bubbles continuing to rise seem to seek the meniscus of surface foam. We arbitrarily set a diameter of clear oil surface of 3/10 in. and timed the settling to that degree of completion. Fig. 18 shows the graduate with a collecting head and handle. Fig. 19 illustrates the settling process as to foam formations and, by alphabet order and increment, roughly indicates the relation of time to configuration of the "beer head."

Such a gaging method is crude, having, for one thing, an error as to initial and final quantities of finely divided air. It did, however, convince us that oil/air mixtures made up by bench rigs to simulate engine sump oil should be checked for fineness of division as well as quantity of air content. Otherwise, bench tests of any separating device, be it tank, centrifuge, or other, can give wrong or misleading answers. We have discontinued development tests of one static and two centrifugal types of separators until we have a better yardstick. We think a modified Westphal specific gravity balance will do the trick, since what needs to be read is the rate of change of density out toward the end of the settling cycle.

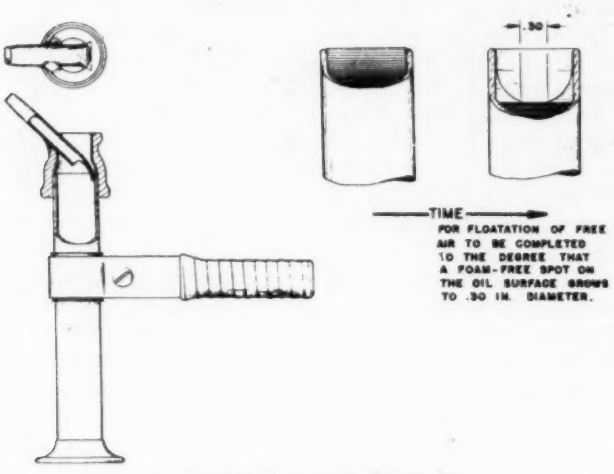
■ Centrifugal Separators

Our work with centrifugal separators was done to provide a possible alternate means of scavenge oil reconditioning in the event that the best that can be done with temperature and circulated volume of oil is not good enough for the worst cases of emulsion.

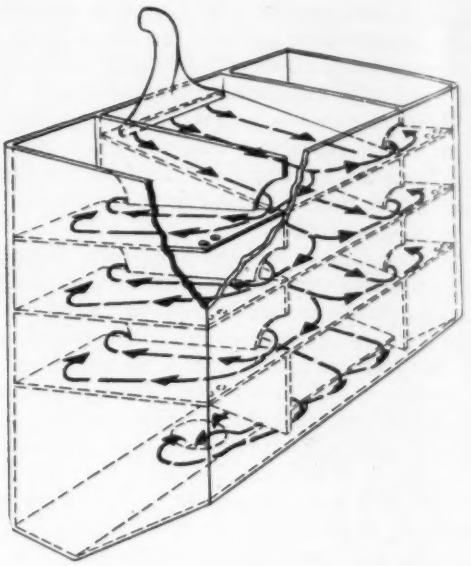
Cooperative work with the Sharples Corp. in separators of the stationary type has indicated definite possibilities in



■ Fig. 17 – Effect of oil supply temperature on engine pressure pump inlet suction


.30
TIME
FOR FLOATATION OF FREE AIR TO BE COMPLETED TO THE DEGREE THAT A FOAM-FREE SPOT ON THE OIL SURFACE GROWS TO .30 IN. DIAMETER.

■ Fig. 18 – Scavenge oil sampler



■ Fig. 20 – Schematic view of ramp baffle tank

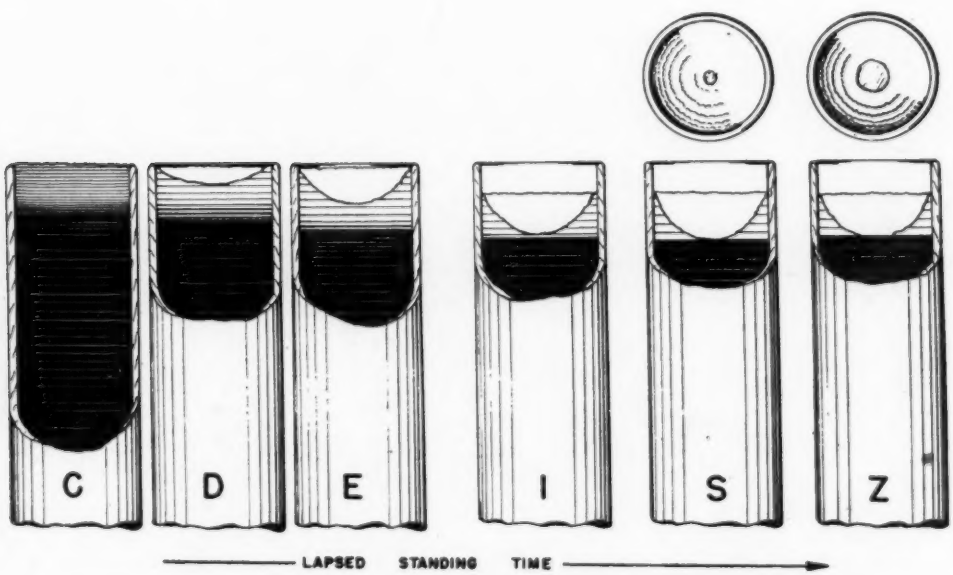
the principle of organized low velocity flow and the take-off of segregated air within the oil tank.

■ Oil Boost Pumps

As to the third item, oil boost pumps, it was recognized that this device offers an expedient to offset or alleviate a case of high air entrainment, when it is found to limit service operations, other "fixes" not being found feasible. We were primarily interested in the potential usefulness of such a pump, oil tank mounted, to raise the ceiling of oil systems of frozen production design, but with the other possibility taken into account it seemed worth some study. In cooperation with Ingersoll-Rand Co. a design was developed which clears the obstacle of cold-starting, high power absorption, and attendant excessive weight. For an engine having an oil flow rate of 200 lb per min the pump provides 3 psi boost up to 40,000 ft with a 0.25-hp electric motor and, if it is desired, it can be arranged to come into action only above a predetermined altitude. It would make 35,000 ft seem only 21,000 to the engine pump and, at that altitude, reduce a 15% entrained air volume to an effective 8%. The complete unit would weigh about 7 lb.

■ Survey Results

Our survey of oil supply temperature practice in the field had its incentive, naturally, in the fact that oil pressure troubles with which we had contact were, in the majority of cases, with installations operating at the lower of commonly accepted temperatures. Our own latest headache



■ Fig. 19 – Settling of a scavenge oil sample

from entrained air was a very active airplane model in which the operating oil temperatures were quite consistently below 160 F, and as low as 140 F was allowed.

The survey results showed that, in this year's high-production airplanes inlet oil temperatures of 180 F or above are exceptional. Typical in one of the two largest bombers is an accepted range of 158-169 F and in the other 158 F was found to be representative except in summer. In several single-engine models used in large number the accepted range is 140-158 F. By far a large majority of all aircraft reviewed are shown to operate with oil supply at or below 176 F.

In the survey we also asked that reasons for this be solicited and objections to temperatures of 180-190 F invited. On these two points we drew complete blanks.

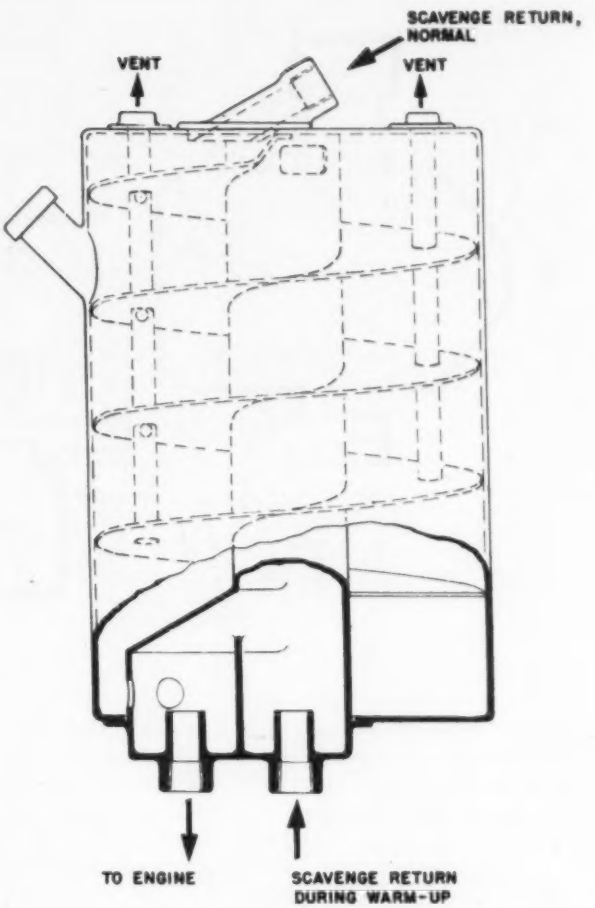
We then approached the oil cooler manufacturers on those subjects. They were able to throw some light on the matter of widespread acceptance of low oil temperatures. We wish to thank them and hope that our brief interpretation meets with their agreement.

We understand that a widely used type of thermostatic element has the natural characteristic of regulating oil temperature at a lower level than other types which it has superseded although set to close the oil routing valve at the same temperature. It appears that the matter of interchangeability influenced the design of the later element.

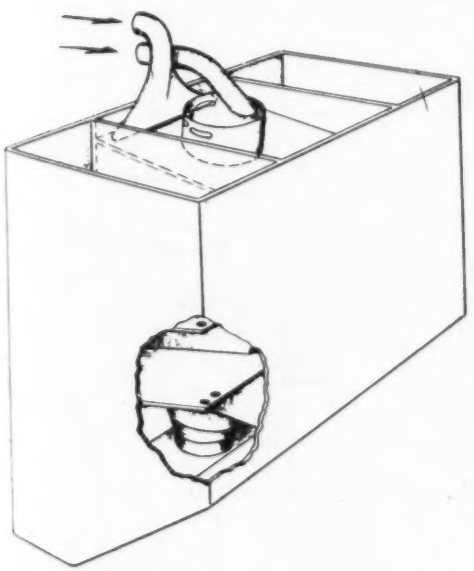
The lower operating range resulting has not caused any obvious trouble, so has been quite generally accepted.

There has been no objection offered from any source to a general use of a temperature control range of 180-190 F for normal operation. A review of some specifications which have directly influenced the designed control range of various oil temperature units for the past two years is interesting. In 1943 a widely used automatic oil cooler was intended to control its outlet to 170-175 F. In 1944 two models in large production were set to start closing

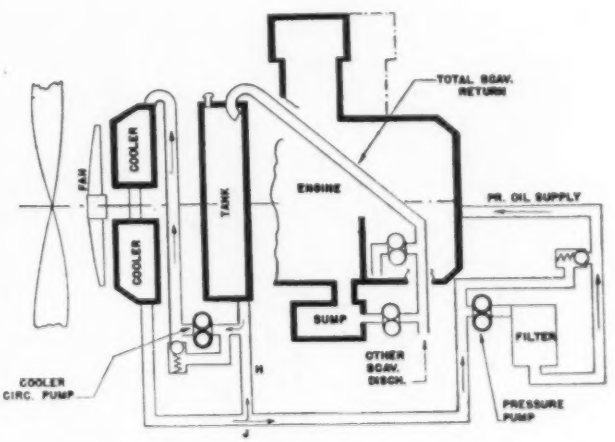
[3] See Fig. 2 of Gulf Research & Development Co. Report KG OO(41), Oct. 27, 1944: "Methods of Minimizing Entrained Air in Aircraft Lubrication Systems," by G. S. Peterson and B. P. Walsh.



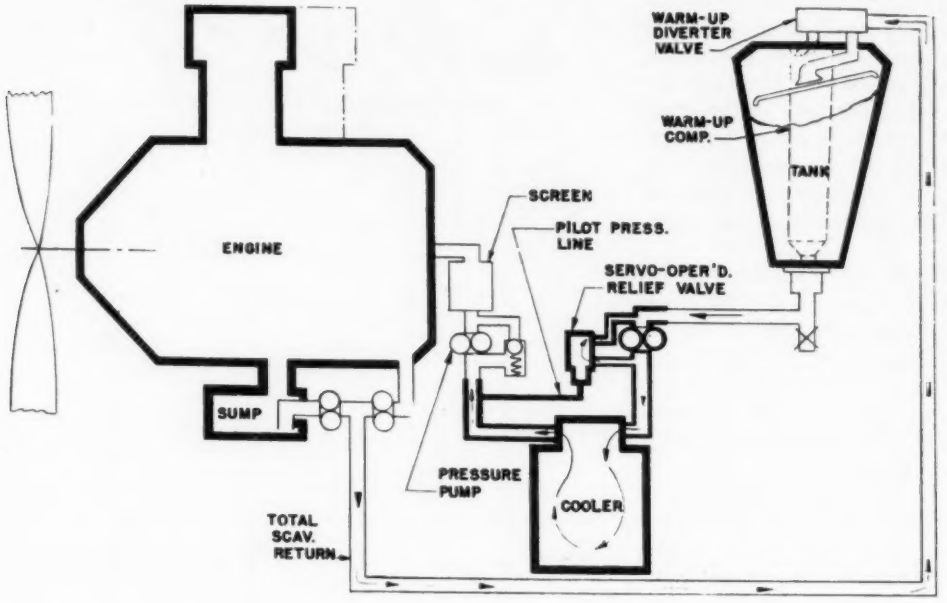
■ Fig. 22 – Long-route, low-velocity tank[3]



■ Fig. 21 – Ramp baffle tank with dilution (warmup) provision



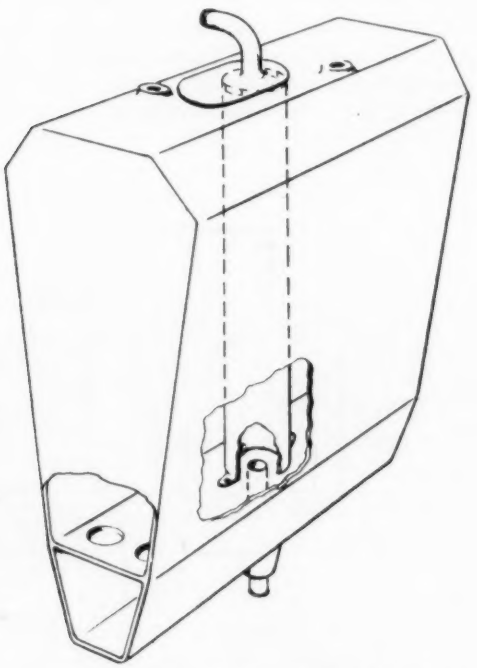
■ Fig. 23 – Schematic diagram of FW-190 oil system



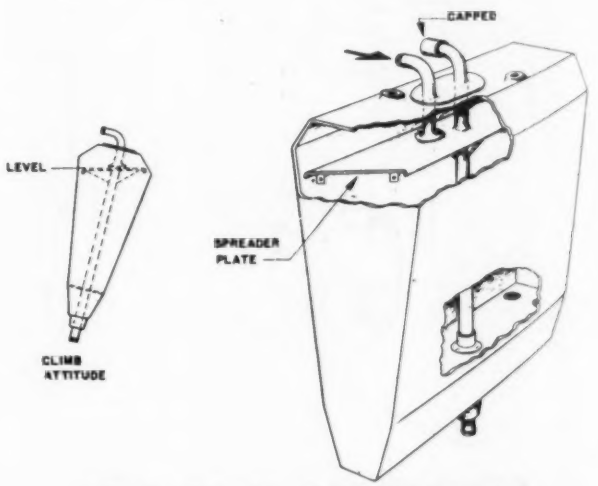
■ Fig. 24 – Schematic diagram of hot tank oil system

the bypass at 130 F and fully close it at 160 F. In 1945 a popular cooler limited the bypass valve range to 143 F start of closing – 153 F full closed. Within the past two years some automatic controls have been produced which have a range of 158-167 F and others a range of 167-176 F and this may be indicative of a trend toward removal of a definite handicap to deaeration.

The time factor in the deaeration process, as established by the routing of oil through a tank, is so tangible a thing to work with that there is quite a variety of ways to modify it. If provisions are made in a tank for long routing and low velocity it follows almost inherently that some of the means provided may be above oil level and offer dispersion surface for entrained air. Fig. 20 shows, schematically, a tank in which an extremely long-route, low-velocity path is provided by a series of ramp baffles. Flight test showed it to be an excellent separator at 180 F. Fig. 21 shows how simply a dilution and quick warmup compartment can be adapted. Fig. 22 is another form of long-route, low-velocity tank also adapted to dilution needs. This is Gulf Research & Development Co.'s conception of



■ Fig. 25 – Hopper type of oil tank (flight test airplane)



■ Fig. 26 – Oil tank modified for hot tank system

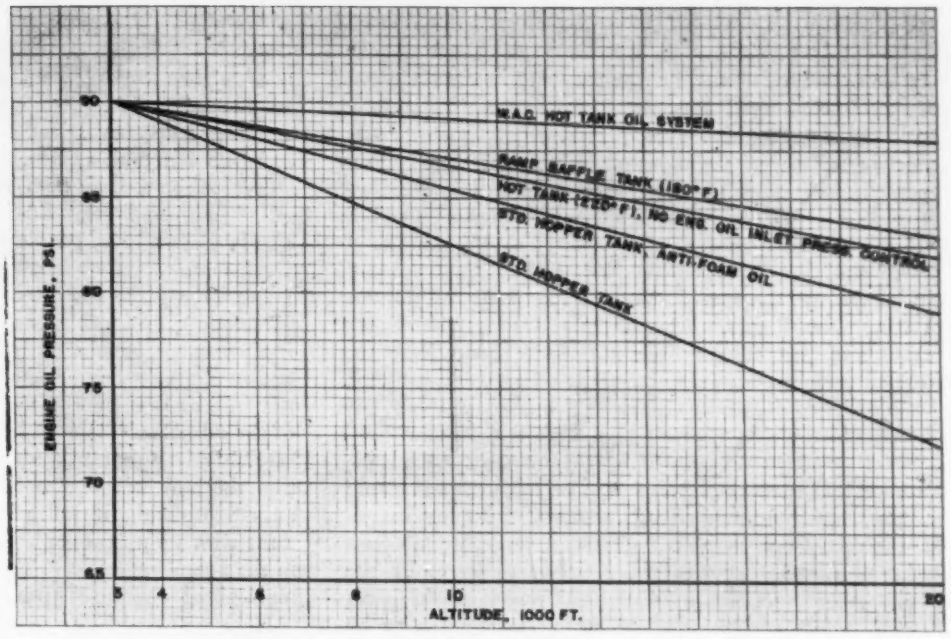


Fig. 27—Oil pressure drop in climb to 20,000 ft for various oil systems

a long-dwell tank and has shown good performance on the bench.

This principle of oil tank function has been extensively studied and developed by United Aircraft Products Corp. and their present project rates the serious consideration of the aircraft industry as a whole.

In the early months of 1942 Germany had Focke-Wulf-190's and Dornier-217's, both powered with BMW-801A engines, operating with scavenge oil routed, hot and direct, to the tank and from there through the oil coolers, by a pump for that purpose, and thence back to engine. *Flight* magazine[4] wrote up the oil system installations so soon thereafter that it seemed that the British had simply told a good fighter pilot to go out and get one or two of each for engineering reference if nothing else. They may have been influenced by the fact that some Spitfires had been attacked from above by FW-190's while flying at 40,000 ft.

The basic principle of the oil system was seen at once to be sound. Tests in deaeration which Gulf had made for us had proved that repeatedly. Fig. 23 reviews the FW-190 system schematically and omits some details extraneous to the matters of deaeration and pressure pump supply. We have been told that the original reason for the disposition of oil tank and cooler and the addition of a pump to force oil through the cooler was not a matter of improved deaeration but one of high flow resistance of the particular cooler used being objectionable as to scavenge back pressure. It is known that the relief valve on the cooler circulating pump is set to about 180 psi.

It is interesting to see that the pressure pump supply line from the coolers is branched at *J* to join the tank and circulating pump inlet through line *H*. From authentic German drawings we find out that the pressure pump and the circulating pump have the same size and speed. How-

[4] See *Flight*, Vol. 41, June 25, 1942, pp. 635-639: "Dornier-217E"; Vol. 42, Aug. 13 and 20, 1942, pp. 169-173, 201-202: "BMW-801A," by F. C. Sheffield.

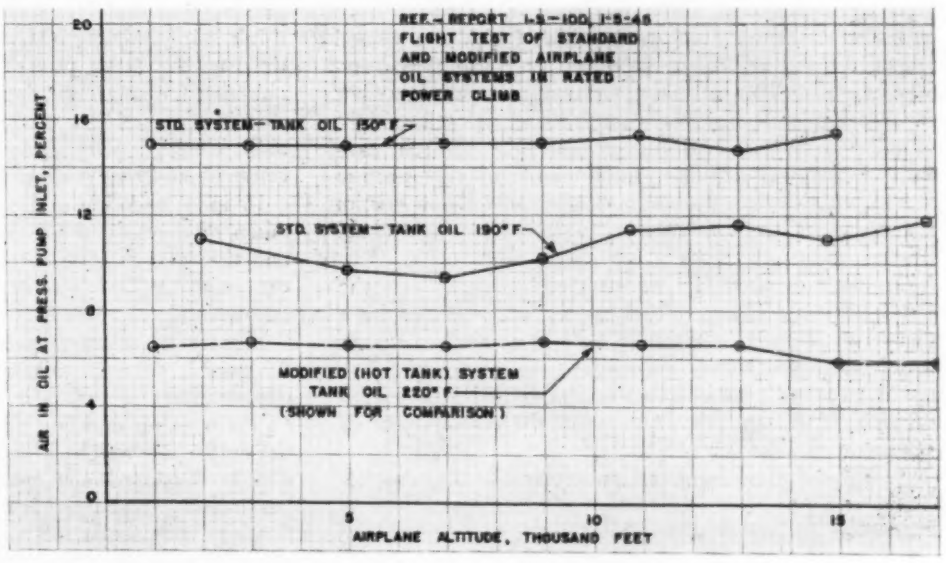


Fig. 28—Effect of tank oil temperature on aeration at pressure pump inlet

ever, at any altitude below the capacity ceiling of the pressure pump it shunts some of its flow through its relief valve, while the circulating pump operates at full capacity. So, at normal conditions there is some recirculation of cooled oil by way of line *H*, plus some due to the difference in discharge pressures at which the two pumps operate. Line *H* must have better justification and we propose it is to provide better pressure pump supply in winter starting than congealed coolers would permit.

Our study of the system focused on line *H* and we concluded that we did not like it. For one thing, oil dilution practice should make it unnecessary. For another, the pressure of this line nullifies a pressure boost from the circulating pump which is potentially offered. As it stands, the higher flow rate of that pump cannot impose any pressure on the oil pressure pump because the useful head is never greater than offered by the tank oil level. We reasoned that if an extra pump is to be considered it would be well to take more advantage of it and, further, to regulate the advantage, namely, the boost it can offer the pressure pump so that engine oil pressure as shown at the cockpit gage will be as constant as possible. If the boost pressure available from the circulating pump was to be regulated, the logical means was a relief valve to shunt its flow. The latter would also avoid excessive circulation through the cooler. But, since the oil cooler offers a variable flow resistance between the circulating pump and the engine pressure pump, a relief valve alone on the former would not insure very uniform boost. So, in our own version of the hot tank system this relief valve was arranged to move against its spring in response to the pressure in the engine oil inlet line at the pressure pump. In Fig. 24 you will see the pilot line which transmits a low order of positive pressure to the piston end of the circulating pump relief valve. The oil tank is shown provided with a compartment for dilution and a warmup as is. One or another type of thermostatic valve would divert the entire scavenge oil flow after warmup to the main section in any simple manner that would cause it to enter the tank oil at very low velocity. We show a simple spreading baffle. Actually our flight test installation omitted the dilution and quick warmup provisions as all the deaeration investigations were made in warm weather.

As mentioned earlier, an airplane was available which, in standard oil system configuration had a prime case of high air entrainment. The engine was a two-row type with the complement of gears which includes a geared propeller shaft, a two-speed supercharger and three scavenge pumps. The oil tank was of the hopper type but the scavenge discharge was centered and pointed straight down in the hopper head. Fig. 25 shows the tank schematically. There was a diving compartment and valves at the bottom but all valves were removed once the characteristics of the standard configuration as to deaeration were established.

Fig. 26 illustrates the modifications made to test the WAC hot tank system. The hopper was replaced by a tube strut to stiffen a spreader plate positioned so that it would be approximately horizontal in normal climb. A special scavenge return fitting was added and shaped to spread its discharge over as much of the plate surface as possible.

The investigation flight test programs included the following items:

Standard oil system.

Standard oil system, oil supply treated with antifoam agent.

Hot tank arrangement but with ambient pressure at engine inlet.

Ramp baffle tank.

WAC hot tank system.

Wright-Gulf indicators were not available for the test of all of these items. Engine oil pressure was taken as a good criterion of oil condition in regard to air content as all work was done at or below 20,000 ft and we believe a fair general comparison of the five items is presented by Fig. 27. Here the drop in engine oil pressure from initial setting is shown by an arbitrarily drawn straight line for rated power climb to about 20,000 ft. The loss of pressure with the standard oil system was 18 psi. Antifoam treatment of the oil helped considerably, showing an 11-psi loss. The reverse relation of tank and oil cooler in the system, so that the tank ran at 220 F resulted in a pressure loss of only 8 psi and the ramp baffle tank was practically the same with a 7-psi drop. You will note that oil temperature in the latter was 180 F. The complete hot tank oil system maintained setting pressure within 2 psi.

Now we would like to revert to the point we can call the major theme and show flight test data which isolate the effect of oil tank temperature. Fig. 28 is a plot of air percentages read with the W-G indicator with oil tank temperature held at 150 and 190 F, respectively, with the standard oil system in two climbs and, for reference, readings taken from the WAC hot tank installation in a similar climb. Bearing in mind that the standard configuration of hopper tank was a quite radical case of undamped, concentrated scavenge return discharge, please note the improvement in deaeration that resulted from an increase of 40 F in tank temperature. The improvement is nearly half of that accomplished by a major change in oil system configuration as represented by the developed hot tank system.

In conclusion it is submitted that the laboratory, bench, and flight test results we have reviewed make it obvious that oil tanks should be operated at as high a level of temperature as engine specifications name so that air separation is not handicapped.

We hope it is just as obvious that we cannot expect to get much separation of air if scavenge oil is constantly routed through a warmup compartment, downward, at a rate a good deal faster than fine bubbles can rise. We think it is amply demonstrated that the lowest flow rate of the largest possible part of the reserve oil is the obvious objective for new tank design.

It is proposed that the emulsifying process in modern aircraft engines is inherent to the kind of mechanism they comprise and that we face that fact squarely.

If, in new design and present operations, advantage is taken of the natural behaviour of scavenge oil when hot and thin and moving through a vented space slowly we can expect to find it unnecessary, in most cases, to carry a power-driven separator.

If scavenge mixtures persist in retaining too much air for good pressure regulation although given a reasonable opportunity to separate, it may become necessary to add an extra pump to adapt the hot tank or to spend about the equivalent weight in a centrifuge. The hot tank system is developed and ready. Final refinement of the power-driven centrifuge is not far off, awaiting only *justification* for its use.

PACKAGE HEATER DEVELOPED For BOEING STRATOLINER

by HAROLD R. PORTER*
Standard Oil Co. of Calif.

■ Fig. 1 – Boeing SA-307B-1 in flight

AIRPLANE heating systems became necessary with the development of airplanes with enclosed passenger quarters. Various methods for the heating of airplane cabins have been developed and, interestingly enough, in the author's knowledge all types of systems ever developed may be found on airplanes flying today. Following is a listing of the various types of airplane heating systems:

1. Exhaust shroud.
2. Exhaust intensifier tube.
3. Exhaust exchanger with secondary exchanger.
4. Exhaust boiler with radiator (water system and glycol system).
5. Combustion heater (engine connected and with fuel supply independent of the engines).

All of the above heating systems have serious drawbacks. The fact that they are all still in use indicates that the industry has made little progress in the development of this very important airplane accessory. This condition has led to the establishment of an engineering laboratory by TWA in which developments are carried out on heating systems as well as on other airplane accessories.

The return of Boeing 307B-1 airplanes to TWA for conversion to passenger service presented a practical opportunity to make a development in aircraft heating systems. This came about since the heating system as installed was totally inadequate and since no equipment already developed was available which would meet the company specifications for a satisfactory airplane heating system. Fig. 1 shows the Boeing Stratoliner.

The package heater described below was the result of a cooperative development by TWA, Surface Combustion, and Minneapolis-Honeywell companies.

■ Design Objectives

Following is a listing of the objectives on which the design of the package heater and method of admitting heat to the Boeing 307B-1 airplane have been based.

1. A maximum degree of passenger comfort must be attained. This means that:

a. The ambient air temperature distribution must be uniform and at a comfortable level in all parts of the cabin and at all times with the airplane in flight or with the airplane on the ground with the engines stopped.

b. Radiation effects to cold walls and windows must be eliminated.

c. An adequate supply of fresh air free of odors must be provided.

d. Undue drying effects due to high-temperature and low-humidity air must be avoided.

e. There must be no drafts.

2. The maximum in reliability of the heating system must be achieved. To accomplish this:

a. The heat source must consist of two independent components which may be operated singly or together and arranged such that the failure of one component does not affect the other component.

b. The component parts must be, in so far as possible, only those which have been tried and proved in airplane service.

c. The heating system must be simple to operate.

3. The maximum in ease of maintenance and repair must be obtained. To attain this:

a. All elements of the heat source must be incorporated in one package which may be readily replaced in the airplane and which may be bench tested in its entirety.

b. Component parts must be easily available for servicing and so far as possible should be mounted in groups which may be tested as an assembly.

c. Special attention should be paid to time-saving features, such as quick-change clamps and fasteners.

d. The heater packages must be interchangeable between airplanes.

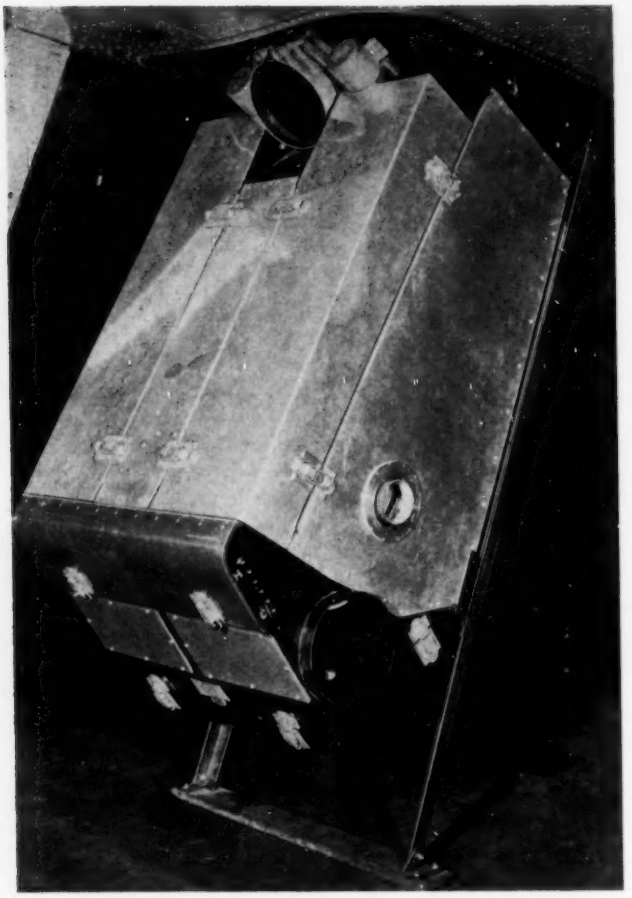
4. The maximum in safety of operation must be achieved. To attain this:

DESCRIBED here in detail is the package heater developed for the Boeing Stratoliner as the result of cooperative efforts between TWA and two other companies.

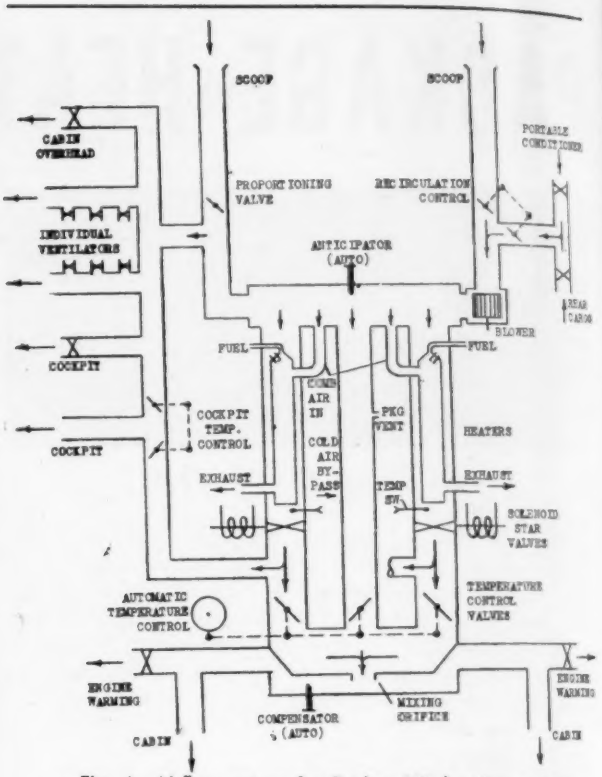
THE AUTHOR: HAROLD R. PORTER (M '37), senior product engineer in the product acceptance department of Standard Oil Co. of Calif., joined that organization upon graduating from the University of California in 1930. His one other commercial affiliation was with Transcontinental & Western Air, Inc., where he worked from 1942 to 1945 developing aircraft heater systems.

[This paper was presented at a meeting of the Kansas City Section of the SAE, Kansas City, Mo., June 7, 1945.]

* Paper presented while author was senior engineer with Transcontinental & Western Air, Inc.



■ Fig. 2 – Completed package heater on installation rack



■ Fig. 4 – Airflow system for Boeing 307 heating system

a. The heater must automatically shut off and remain off in the event that a critical operating condition arises.

b. The heater must be designed so that it is inherently safe even though a safety feature should fail at a critical time. Also, all parts must be arranged to "fail safe."

c. The heater package must have positive and controlled ventilation, must have an adequate fire warning system and fire extinguishing means, and it must be possible to shut off all air supply to the heaters in an emergency.

d. Adequate heat must be available for warming the cockpit at all times and its control must be totally independent of the control of heat to the cabin.

e. Heat must be available for defogging and de-icing the cockpit windows at all times.

f. The electrical parts on the heater must be protected with adequate circuit breakers.

g. The operation of the heating system must entail a minimum of attention from the flight crew. Also, its operation must not interfere in any way with the airplane performance.

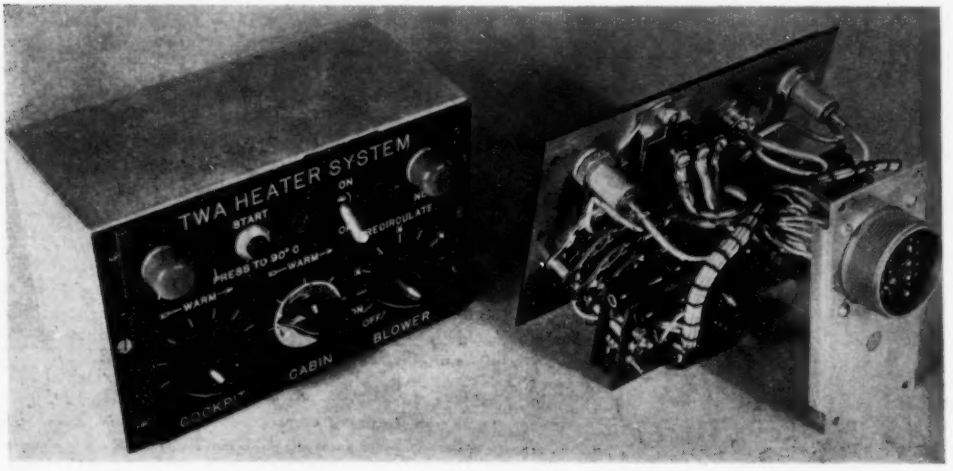
5. Miscellaneous objectives to be achieved are:

a. The heater must be light in weight and require a minimum of space.

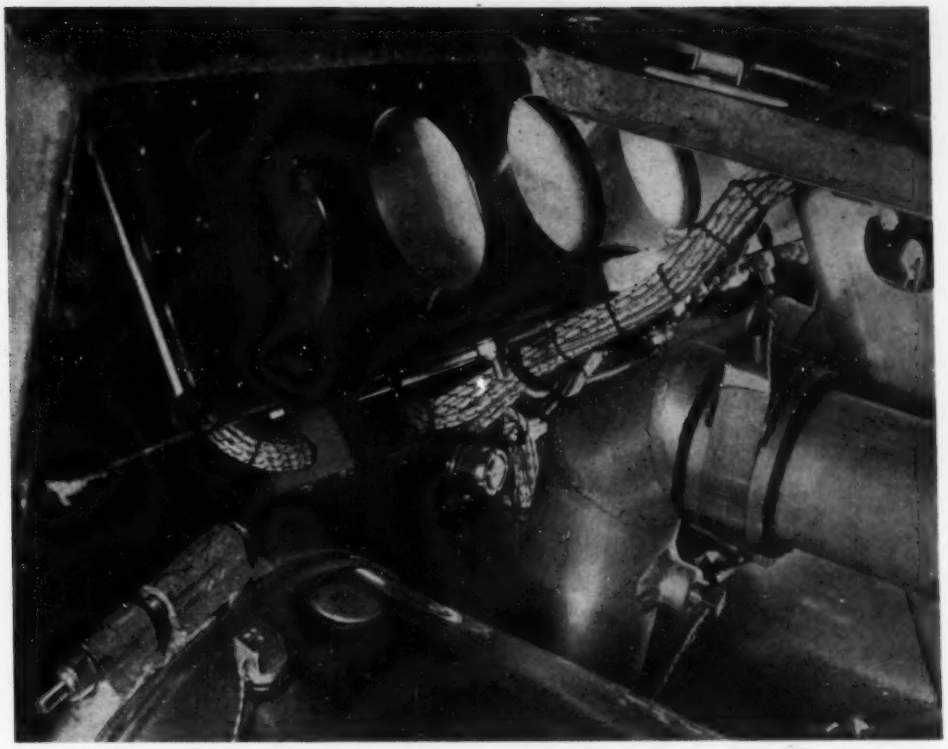
b. The heating system must operate without perceptible noise, vibration, or other annoyance.

■ Description of Package Heater

Fig. 2 shows a completed package heater. Fig. 3 shows a completed package control box.



■ Fig. 3 – Package control box



■ Fig. 5 – Heater package with covers removed (note CO_2 nozzle)

The basic design of the package heater is such that the inlet air passes through the combustion heater units and also through a duct bypassing these units. The air leaving the heater units is held at 200-220 F under all conditions. Temperature modulation is obtained by means of face and bypass dampers which proportion the relative amounts of heated and bypassed air. These dampers are actuated by the automatic temperature control. The heated air supply to the cockpit is taken ahead of the control dampers and in consequence is independent of damper setting. Fig. 4 illustrates the air paths through the package. Fig. 5 shows the heater package with covers removed.

Following are detailed descriptions of the component parts and systems incorporated in the package heater:

Combustion Heaters – The heater units were developed especially for this application by Surface Combustion. They operate on gasoline drawn directly from the airplane tank and are ignited by a 400-c, 6000-8000-v transformer. Each heater unit has a capacity of 115,000 Btu per hr. Fig. 6 shows one of the Janitrol heaters.

Fuel Supply System – Fig. 7 shows the fuel system. It will be noted that this system has one main gasoline filter for the package and that each heater unit has its own pump, nozzle filter, pressure gage, fuel regulator, relief valve, and fuel solenoid valve. All these are grouped together and are located in a separate compartment in the heater package.

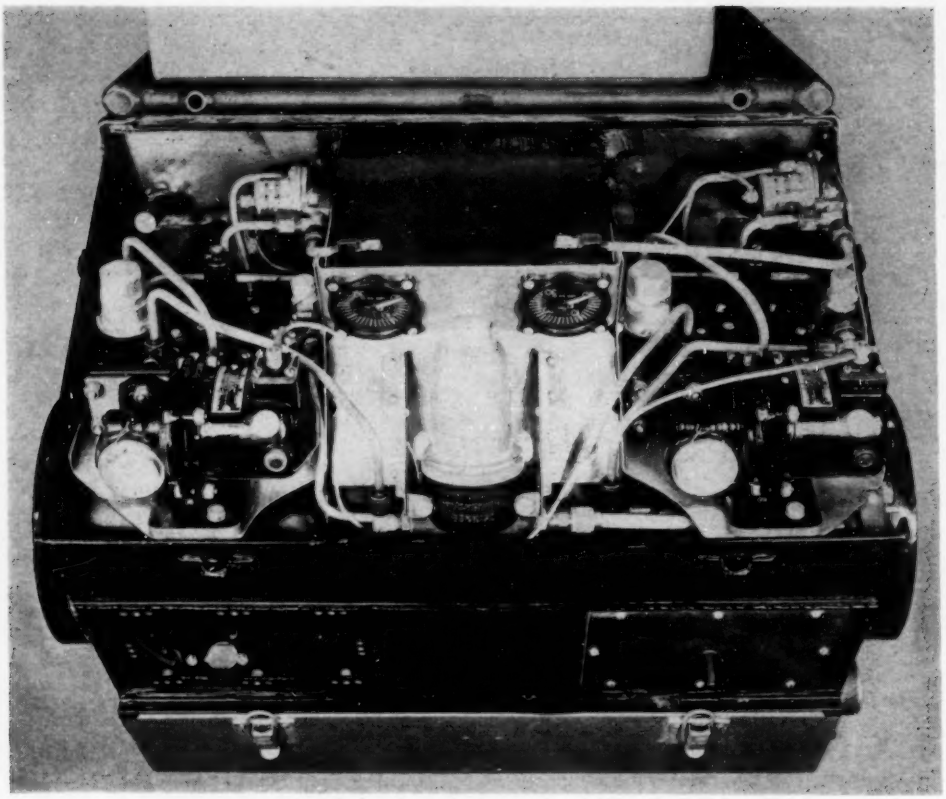
Combustion Air Supply System – The combustion air supply is taken from the bypass duct through a ¼-hp motor blower which is provided for ground operation or for recirculation in flight. No air pressure regulators are used.

Exhaust System – The heater exhaust gases are passed overboard through a stainless-steel pipe containing a section of flexible stainless steel tube. The flexible section has a stainless-steel interliner. Ventilating air from the package is passed between the exhaust pipe and a shroud about it and then between the interliner and its flexible section. After passing the interliner, the ventilating air mixes with the exhaust gases and serves to cool them. Fig. 8 shows an exhaust pipe assembly. Fig. 9 shows the exhaust stacks through the belly of the ship.

Fire Prevention System – Each airplane is provided with a 1.2-lb CO_2 bottle which is located and controlled at the flight engineer's station. This bottle is connected to two nozzles in the package, one which discharges into the fuel accessory compartment and the other into the main accessory section of the package. The nozzle, discharging into the main accessory section, may be seen in Fig. 5.

Electrical System – Fig. 10 shows a schematic of the electrical system and indicates the equipment used. This system consists in essence of a control box, a circuit breaker panel, electrical relays, a low temperature limiting thermo-

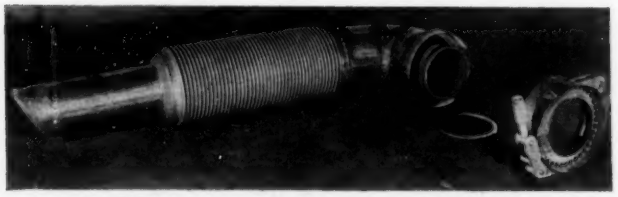
■ Fig. 6 – Janitrol heater unit



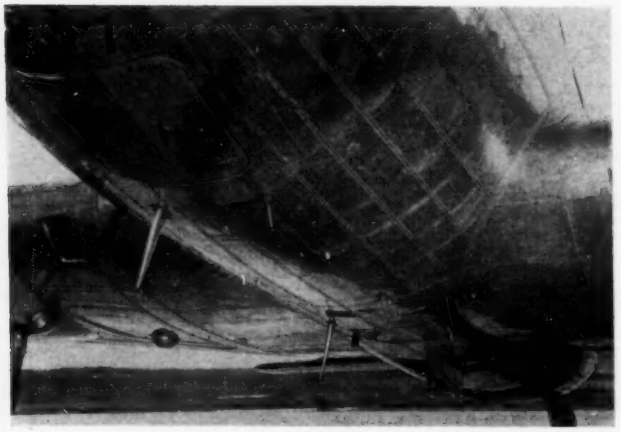
■ Fig. 7 – Package heater fuel system

switch, a control thermoswitch, a high temperature limiting thermoswitch, fire detectors, a ram pressure switch, and a solenoid for actuating the star valves in the heater air stream.

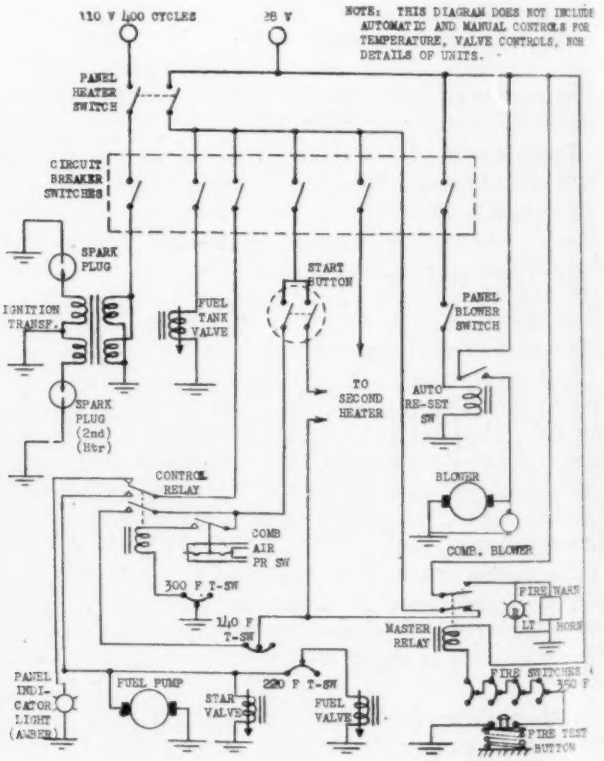
The principal features of the electrical system are the method of controlling the heaters and its lock-out provisions. Control of the heaters is obtained by a thermoswitch in series with the fuel supply solenoid. This thermoswitch is normally closed and is set to open at 220 F. Experience has shown that this arrangement will hold the heater off temperature within ±10 F in any



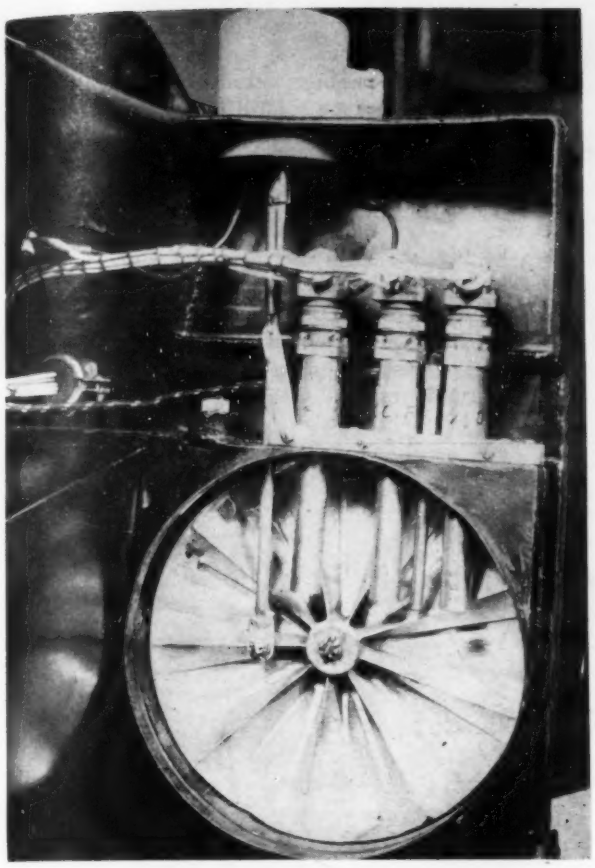
■ Fig. 8 – Package heater exhaust assembly



■ Fig. 9 – Location of heater exhaust stacks on Boeing 307B-1



■ Fig. 10 – Electrical diagram for Boeing 307 heating system

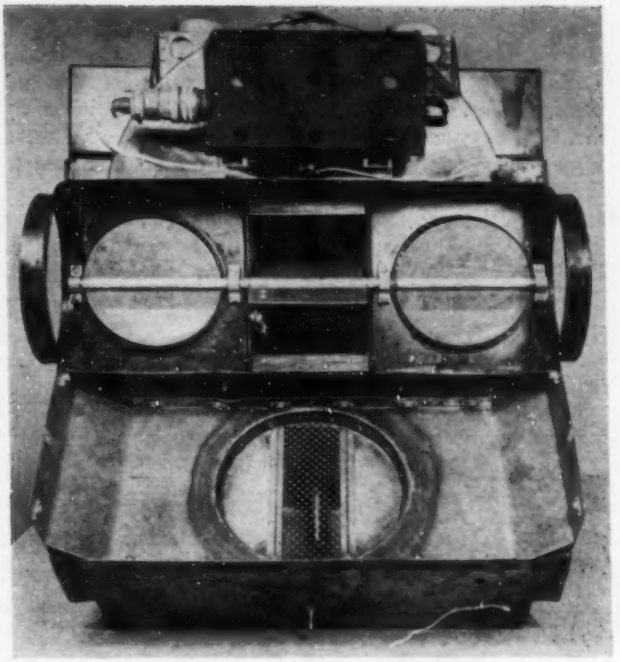


■ Fig. 11 – Location of thermal switches in star valve

damper valve position. At ½ output, lag in the system is such that the heater cycles for 5 sec on and 5 sec off.

The safety limits imposed by the lock-out circuit and

■ Fig. 12 – Junction box and electrical accessories



■ Fig. 13 – Amplifier and air control dampers

which, if exceeded, will completely shut down the heater unit in question are as follows:

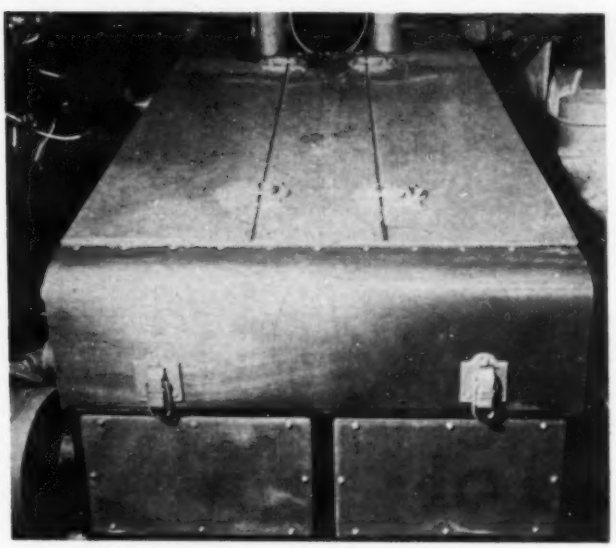
1. Low-pressure limit on combustion air – to protect against failure of combustion air pressure.

2. Low-temperature limit – to protect against failure of spark-plug or any other failure which would drop the heater output to less than 100 F.

3. High-temperature limit – to protect against failure of heater control or any other failure which would increase the air-off temperature to more than 350 F.

In case any one of the fire detectors should become exposed to over 350 F, both heaters will shut down completely.

It should be noted that when a heater is termed completely shut down, the fuel pump stops and the star valve restricts all passage of air through the heater. Fig.



■ Fig. 14 – Package heater installed on Boeing 307B-1 airplane

11 shows the method of locating the thermoswitches and star valves.

Automatic Control – The automatic control equipment consists of an anticipator ductstat located in the incoming air, a compensator ductstat in the air off the heater, a cabinstat on the forward bulkhead, a motor connected to the air control dampers, an amplifier, a junction box, and an electrical override switch. This control equipment was developed by Minneapolis-Honeywell for this application. Fig. 12 shows the location of the junction box along with the other electrical accessories. Fig. 13 shows the amplifier and the air control dampers.

Mounting of Package Heater and Package Control Box in Airplane – Fig. 14 shows the package heater mounted in the airplane. The quick-acting clamps for attaching to the ducts should be noted. It has been found easily possible to remove one package and to replace it with another in 20 min. The package control may be changed in the same time required for a heater change.

■ Ducting and Radiant Wall Installation

A view of the ducting installation with airflows is shown by Fig. 15. Also shown in this latter diagram is a view of the control panel with a listing of the motorized valve sequences controlled by the "recirculate" switch. Following is a description of the airflows through the system and corresponding valve settings under various operating conditions.

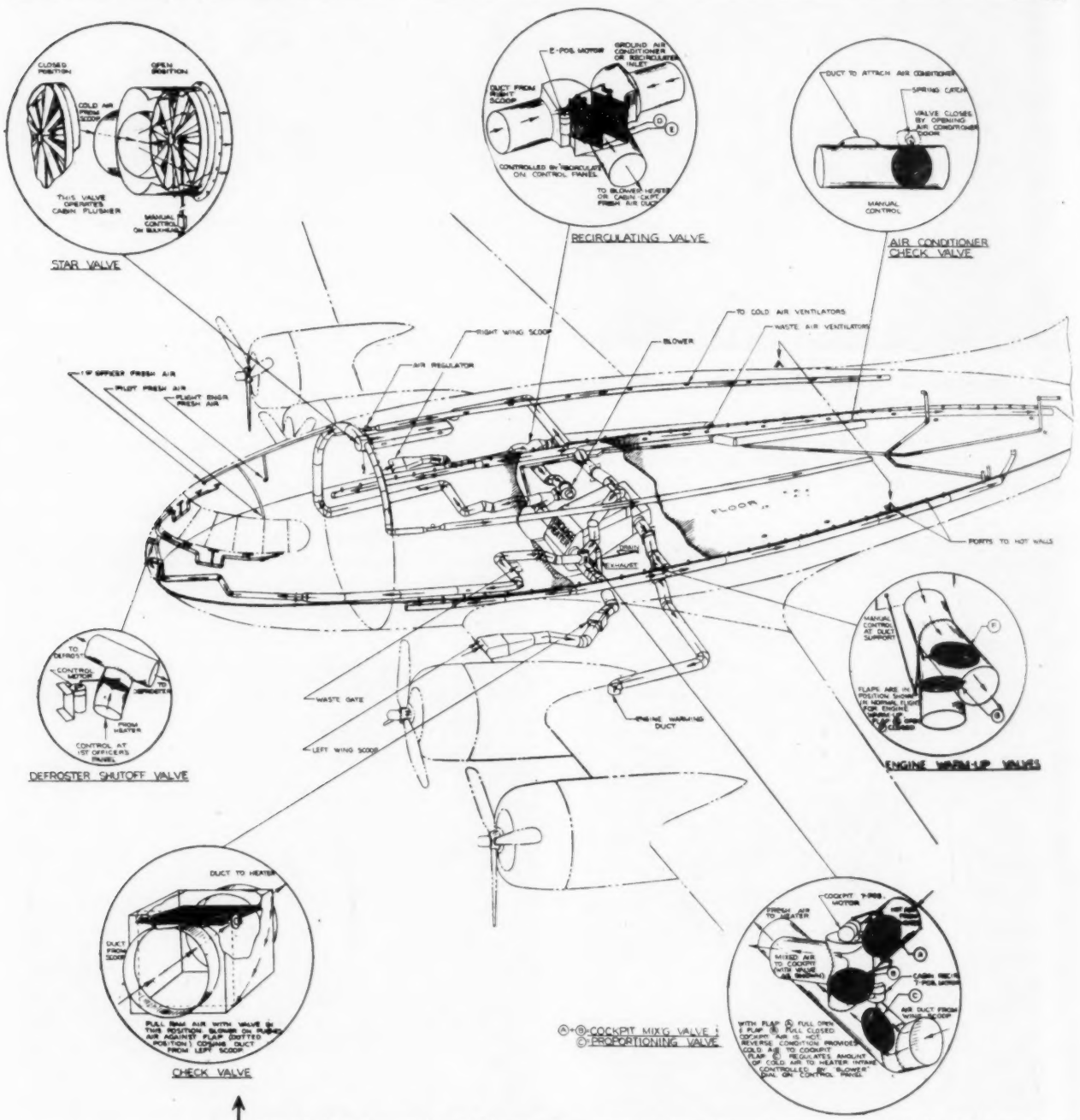
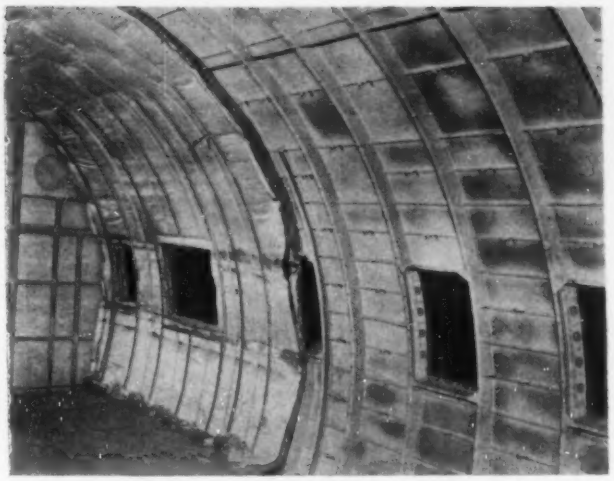


Fig. 15 – Heating and ventilating duct system – condition shown; heat on, normal flight, full ram air →

Full Ram Air – Air is admitted through both scoops when operating on full ram air supply. That entering the left scoop passes through a check valve, then the proportioning valve which is open, then a cockpit mixing valve where some is directed to the cockpit, and finally enters the heater. The air entering the right scoop passes through the recirculating air valve, then through the ventilating air blower which is not operating, then through a tee where some is diverted to the cold air ventilating system, and finally enters the heater. On passing through the package, the air is heated. That directed into the cabin distributing ducts is modulated to meet the cabin demand and that directed to the cockpit mixing valve is maintained at the heater off temperature of 200 to 220 F. After passing through the cabin the air is directed through holes in the floor connecting with the baggage compartments and is passed overboard through the waste gate opening in the accessory section of the airplane.



■ Fig. 16 – Method of installation of radiant walls

Auxiliary ventilation for quick flushing of the cabin may be provided by opening the star valve in the overhead ventilating air duct. Ventilation for the cockpit is obtained by setting the mixing valve control to "cold," and if more air is desired, by opening the defroster valve.

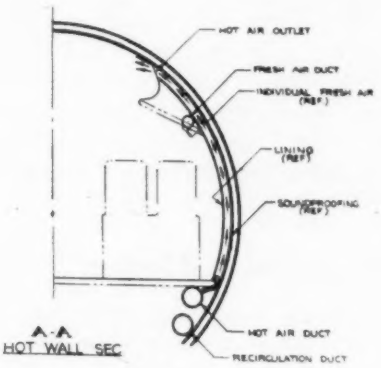
Full Recirculation – No air is admitted from the scoops during full recirculation. The left scoop is shut off by the proportioning valve and the right scoop by the recirculating valve. In this condition air is drawn from the rear part of the airplane through the air conditioner valve which is open and passed through the blower which is operating and enters the right side of the heater. The passage of the air through the rest of the system is the same as when operating on ram pressure with the exception that the only air passed overboard is that which leaks into the airplane.

Partial Recirculation – Partial recirculation is obtained by opening the proportioning valve in the left scoop to an intermediate position with the other parts of the system in exactly the same configuration as for full recirculation. Increase in the amount of recirculated air is obtained as the proportioning valve is closed. The proportioning motor which actuates this valve assumes a definite setting for each position of the "recirculate" switch on the flight engineer's control panel.

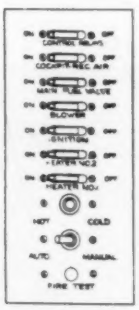
Ground – Ground operation may be accomplished with full recirculation or with air drawn from the right-hand scoop only. This latter condition may be obtained by simply turning the recirculate switch into the "ground" position, in which case the recirculating valve opens the blower inlet to the scoop and the proportioning valve closes.

When operating with a conditioner truck the system is set for recirculation. However, the act of opening the access door in order to connect the air conditioner hose serves to shut off the duct communicating with the rear of the airplane and thereby places the blower in series with the conditioner truck.

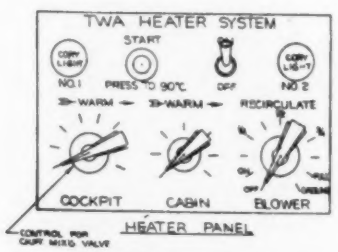
Fig. 16 shows the construction of the radiant wall and illustrates the manner of installing the BC sound deadener and the XMPF insulation. Fig. 17 shows a schematic



A-A
HOT WALL SEC



SWITCH PANEL



HEATER PANEL

EXPLANATION OF "BLOWER" DIAL

1. OFF POSITION - RAM AIR ONLY - (C) OPEN, (D) CLOSED, (E) OPEN
2. ON POSITION - VALVES SAME AS OFF POS., BLOWER RUNNING
3. RAD POSITION - (E) OPEN - (D) OPEN - (B) CLOSED
4. ¼ POSITION - (C) ¼ CLOSED - (D) OPEN - (B) CLOSED
5. ½ POSITION - (C) ½ CLOSED - (D) OPEN - (B) CLOSED
6. ¾ POSITION - (C) ¾ CLOSED - (D) OPEN - (B) CLOSED
7. FULL POSITION - (C) FULL CLOSED - (D) OPEN - (B) CLOSED
8. GROUND POSITION - (E) FULL CLOSED - (D) CLOSED - (B) OPEN

Table 1 – Pressures Obtained with Package Heater for 307B-1 Airplane

	Flight Operation				Ground Operation Blower			
	Ram Supplied		Recirculated		Drawing from Left-Hand Scoop		Recirculating	
		100%		100%				
Run No.	1-4	43-3[a]	1-5	43.4a	B-2 (Full)	B-4 (Control)	B-7 (Full)	B-9 (Control)
Pressure Altitude,ft	7000	7000	7000	7000	740	740	740	740
Air Speed TIAS	175	185	175	185	0	0	0	0
Indicated Ram	14.8	17.0	14.8	17.0	0	0	0	0
Pressure into Heater	12.2	13.6	3.5	3.6	3.7	2.8	2.8	2.1
Combustion Air In	4.3	5.7	6.0	6.2	8.0	7.5	7.5	7.1
Exhaust Pressure	−4.5	−4.2	−1.0	−1.2	0.3	0.3	0.5	0.5
Δ Combustion Air	8.8	9.9	7.0	7.4	7.7	7.2	7.0	6.7
Drain Pressure	0.4	0.4	0.5	0.4	0	0	0.1	0.1
Pressure Upstream Heater Orifice	10.5	11.6	3.0	3.1	1.7	2.0	1.8	1.6
Pressure Downstream Heater Orifice	−1.2	−0.5	0.4	0.3	−0.1	0	−0.2	0
Δ Pressure Heater Orifice	11.7	12.1	2.6	2.8	1.8	2.0	2.0	1.8
Δ Pressure across Heaters	1.7	2.0	0.5	0.5	2.0	1.8	1.0	0.5
Pressure in Cabin Distributing Ducts	....	1.2	....	0.6	0.1	0.1	0.1	0.1
Pressure Cockpit Orifice	9.6	11.0	2.6	2.9	1.3	1.8	1.4	1.6
Pressure in Overhead	11.0	12.4	4.8	4.9	5.0	4.5	4.0	3.7
Pressure Individual Ducts Ahead Regulator	11.0	12.4	4.8	4.9	5.0	4.5	4.0	3.7
Total Pressure Upstream Exhaust Ventilating Air	−0.3	−0.1	0	0.2	0.2	0.2	0.3	0.3
Total Pressure Downstream Exhaust Ventilating Air	−0.5	−0.3	0	0.1	0.1	0.1	0.2	0.2
Δ Total Pressure Exhaust Ventilating Air	0.2	0.2	0	0.1	0.1	0.1	0.1	0.1
Package Pressure	1.0	1.2	0.5	0.5	0.3	0.3	0.3	0.3
Cabin Pressure Relative to Outside	0	0.7	−4.8	−6.0	0.5	0.5	0	0

[a] Orifice increased by 16% in area.
Note: All pressure in inches of water. All pressures static unless indicated otherwise.
All pressures relative to cabin except as indicated

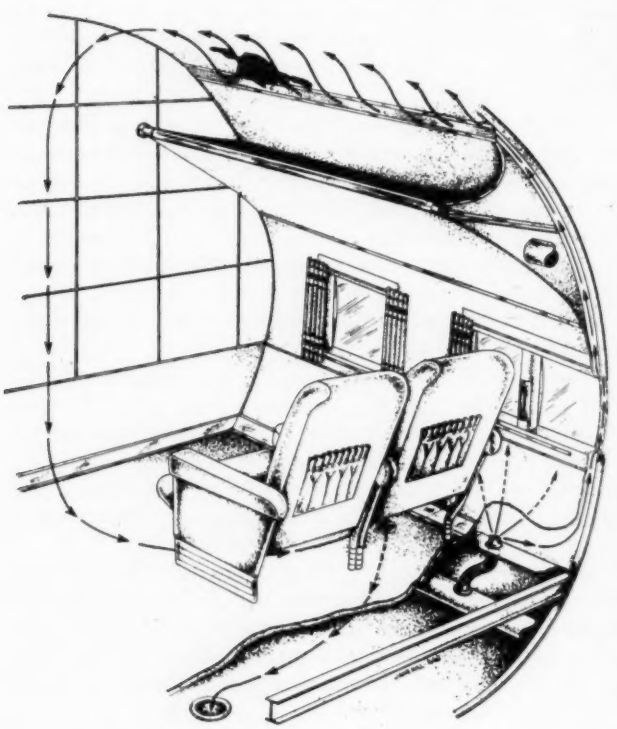
drawing of the wall sections. From this drawing it may be noted that the heated air is admitted to the wall from a large duct with a riser for each circumferential ring. It then passes upward in the space between the inner fabric and the insulation and outward into the cabin from grills located just above each hat rack. In areas containing windows heated air is routed between the double panes and around the window trim. Because of equal pressure throughout the ducts supplying the radiant walls, the rate of airflow through the cabin is very closely the same at any cross-section.

■ Operation of System

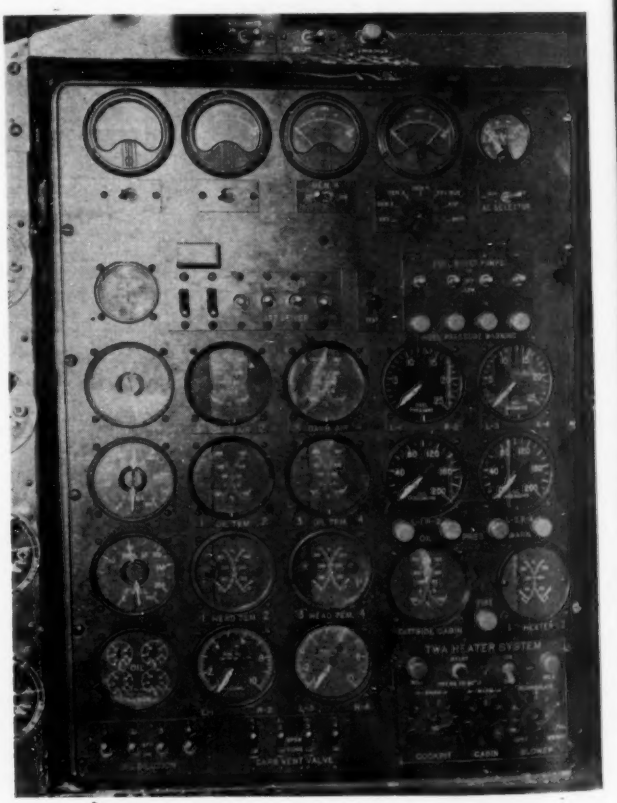
Starting procedure (refer to Fig. 18):

1. Turn all circuit breakers on.

2. Set the "recirculate" switch for the type of operation desired. For ground operation turn to "ground" for in-



■ Fig. 17 – Section through Boeing radiant walls showing airflow



■ Fig. 18 – Flight engineer's panel showing package heater control

flight, and full ram air turn to "off" or for in-flight and recirculated air turn to setting of indicating amount of recirculation desired.

3. Turn main heater control switch to "on."

4. Press "start" button and hold until heater temperatures rise to 90 C. Once this temperature is attained the heater indicator lights will remain off when the "start" button is released.

To operate heaters:

1. Turn cabin temperature control to temperature desired.

2. Adjust cockpit heat to desired level.

To stop – normal:

1. Turn main heater switch to "off."

2. Turn "recirculate" switch to off.

3. Turn circuit breakers off.

Emergency:

1. In case of fire:
   a. Turn main heater switch off.
   b. Turn "recirculate" switch to "full."
   c. Turn all circuit breakers off.
   d. Operate fire extinguishing system.

2. In case the automatic temperature control fails, turn to "manual." Adjust cabin temperature by momentarily holding the override switch to "hot" or "cold" as desired.

■ Preliminary Performance Results

CAA Approvals and Tests – Complete approval on the selection of materials and equipment used in the fabrication of the package heater was obtained from the CAA. These approvals were substantiated both by laboratory and flight tests.

One of the factors most thoroughly investigated was the possible presence of carbon monoxide in the airplane. Extensive tests with sensitive indicators supplied by the Bureau of Standards failed to reveal any carbon monoxide in any flight or ground condition.

Pressures Throughout Heater System – Table 1 shows a resumé of pressures obtained throughout the heating system under several conditions of operation. These do not cover all of the conditions tested but are included herein only as representative values.

Temperatures Throughout Cabin – Complete temperature data were taken on several flights. These data indicated satisfactory temperature distribution and temperature control. Present plans, however, are to repeat these tests under actual service conditions and to use the information along with an evaluation of passenger reaction to radiant walls as the basis for a subsequent paper.

Table 2 – Heat Loads of Cabin and Cockpit Package Heater for Boeing 307B-1 Airplane

	Ground Operation Temperatures not Stabilized Air Drawn from Scoop	Flight Operation Temperatures Stabilized	
		100% Ram	100% Recirculated
Air In, F	41	23	46
Package Out	167	99	107
ΔT Air to Cabin	126	76	61
ΔT Air to Cockpit	95	137	115
ΔP Heater Orifice	1.8	12.1	2.8
Lb Heater Orifice per Hr	1800	4700	2320
ΔP Cockpit Orifice	1.3	11	2.8
Lb Cockpit Orifice per Hr	320	815	430
Lb per Min per Cabin Occupant	0.75	2.0	0.83
Lb per Min per Cockpit Occupant	1.8	4.5	2.4
Btu to Cabin per Hr	54,000	86,000	34,000
Btu to Cockpit per Hr	7000	27,000	10,200
ΔT Cabin over OAT	....	47	44
ΔT Cockpit over OAT	....	49	49
Heat Load Cabin	....	700	770
Heat Load Cockpit	....	350	210
Relative Load Cabin/Cockpit	....	2	3.6

Air Quantities and Heat Loads – Table 2 lists the results of tests made to determine the air quantities passed through the system and the heat loads of the airplane. Under full ram a total of 5600 lb of air per hr was passed through the heater. The airflow to the cabin amounted to 2 lb per min per occupant, while that to the cockpit amounted to 4.5 lb per min per occupant. The airplane heat load under full ram amounted to 1050 Btu per F over OAT per hr as compared to 980 Btu per F over OAT per hr when recirculating. These values are in good agreement. The tests showed further that the cockpit heat load was from approximately one-third to one-half that of the cabin, or that the heat required in the cockpit is a sizable proportion of the total heat required on the airplane.

The evaluation of the Btu capacity of the heaters in terms of airplane heat load indicates that satisfactory heating of the airplane should be obtained at temperatures down to —20 F with full ram and to temperatures considerably lower under recirculation.

EFFECT OF DIESEL FUEL ECONOMY ON HIGH-SPEED TRANSPORTATION

continued from page 157

It is to be expected that a further substantial increase in the life of automotive diesel engines will result from the development and the distribution of equally satisfactory fuels.

In the field of automotive transportation, the diesel engine had firmly established a place for itself before the war.

In spite of stepped-up time schedules, 20% to 40% fuel savings were realized and new high marks established for overall transportation economy.

Under the impetus of the war requirements, a multitude of new applications has been found in related fields where a source of dependable and economical light-weight power was needed.

The unfaltering torque, high flexibility, and outstanding fuel economy of these new powerplants have made friends of many users in the automotive, marine, and industrial fields.

With specific power outputs better than 0.4 bhp per cu in. displacement and specific weights of 8½ lb per bhp, the high-speed diesel engine of today competes with the best of heavy-duty automotive gasoline engines on a weight and power basis.

The unique diesel-engine characteristics of good fuel economy over a wide speed and load range permits the use of larger engines with greater reserve power for acceleration and sustained speed on the hills, this making it possible to haul more payload per day per dollar.

HOW to DRIVE a

EMPLOYERS of truck drivers have an important and sometimes difficult job in training their drivers to conserve the vehicles they use.

One approach to this problem of training drivers to their responsibilities is presented here by Mr. Sinclair, who sets forth and elaborates on a series of "don'ts" that drivers should diligently and persistently practice.

IT is not the purpose of this discourse to indulge in the presentation or discussion of any elaborate driver training program. I am not a psychologist and am therefore incompetent to argue emotional stability. I am neither a phrenologist nor a physiologist and am therefore incompetent to argue mental attitudes, reflexes, reactions, and similar mental and physical characteristics, inhibitions, and complexes. These somewhat disturbing admissions carry with them no pride of ignorance. On the contrary, I have a healthy respect for the advances of science in analyzing the human mind and frame, and the resultant cataloging of individuals as to what they are either capable or incapable of accomplishing. I eagerly anticipate the day when the individual can be scientifically analyzed; and Mr. Brown informed that his particular niche in life is that of an accountant but never a truck driver; and Mr. Jones informed that he is doomed to the life of the truck driver but may never aspire to be an accountant.

Perhaps that day is already here in a fuller degree than we realize. I do not know. However, of one thing I am reasonably certain: that if such services are available they are beyond the economic reach of 85% of employers of truck drivers; for if my statistics are sound, 85% of all commercial vehicles are operated by owners of five vehicles or less. As all commercial vehicles are operated for profit, it seems improbable that professional services of the nature in question are within the realm of sound investment for employers of a relatively limited number of drivers.

Therefore, the great preponderance of truck driver employers – and it is to this group that these remarks are addressed – will have to get along the best they can, resorting to the elemental procedure of (1) assuring themselves that the driver is a reasonably sound physical specimen; (2) acquainting him with his responsibilities; (3) determining by actual tests whether he can fulfil these responsibilities; and (4) by consistent encouragement, incentive, and repetition, endeavor to secure as consistent fulfilment as possible during the course of his employment.

Fortunately for all of us, this discussion is restricted to the second phase of this procedure, namely, driver responsibility in the conservation of automotive equipment. Reduced to its elements, the entire subject matter can perhaps be covered concisely, thoroughly, and in a time-saving manner by merely stating that in order to conserve equipment the driver should diligently and persistently practice the following "don'ts":

(1) Don't start too quickly; (2) don't drive too fast; (3) don't stop too quickly; (4) don't continue to drive when there is some apparent defect, mechanical or otherwise, in the vehicle; (5) don't hit anything; (6) don't let anything hit you; and (7) don't load improperly or excessively.

I am fully cognizant that in setting forth a series of "don'ts" I am possibly running afoul of those who deem a negative admonition a psychological error. However, I don't feel the "don't" approach unwise. I don't feel that in this approach we are building up unhealthy inhibitions in the minds of the drivers. I don't feel the word "don't" causes the auditor immediately to build up resistance to the admonitions. In short, I feel that what is good for the child is good for the adult; and during our entire period of training which continues throughout life, we are more accustomed to prohibitions and are more acclimated to them than any other form of approach. We certainly had adequate training under this approach during wartime.

Seemingly, if driver responsibility can be so briefly defined, there seems to be little to warrant an elaboration. On the other hand, although recognizing the desirability of brevity, we must also recognize that a little knowledge is a dangerous thing and it may follow that inadequate admonitions are subject to wide and varied interpretations. It seems both advisable and desirable to amplify on these elemental "don'ts," so let's explore the potentialities of these admonitions. If the reasons prompting them are understood and appreciated, cooperation in action is more apt than otherwise to follow.

Parenthetically, at this point we emphasize that there is nothing especially original in these amplifications. In a field already so thoroughly covered, there is scant opportunity for originality, even though talent for so doing may be present. As a matter of fact, a considerable portion of the material that follows has been adapted from several excellent and much more comprehensive treatises on the subject, and full credit is given therefor. If our relatively brief condensation of some of this material excites your interest, we strongly urge you to pursue further course of study. Outstanding efforts in this direction are represented by the Driver Manual (White Motor Co.); Wartime Information for Delivery Truck Operators (Studebaker Corp.); The Automobile User's Guide with Wartime Suggestions (General Motors Corp.); Motor-Vehicle Driving Practice (Atlantic Refining Co.); How to Drive a Truck (Timken Axle News); and various similar publications. The material is there and you can adapt much of it to your own peculiar problems and methods of operation.

1. *Don't start too quickly.*

A. *Prior to starting engine.* The first impression is that reference is to the vehicle as a whole, but the initial reference is in reality to preparing the vehicle *prior* to starting. A good truck driver, like a good pilot, should know that

[This paper was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Calif., Oct. 26, 1945.]

a TRUCK

by JAMES W. SINCLAIR
Manager, Automotive Department
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everything is reasonably shipshape before the wheels start to roll. He walks around the vehicle, checking for obvious defects. Even though routine procedures may in some fleets be the responsibility of the shop or mechanical department, he wants to be sure of certain elemental factors, inasmuch as his physical well being, the safety of the general public, and the cargo of the vehicle are his responsibility once the vehicle is under way. What he looks for depends upon the type of equipment he drives, how long it has been in service, and the operation in which it is engaged; but in all cases he checks:

1. For soft or flat tires or uneven tread wear; underinflation under load; and rim and wheel lugs.

2. Steering wheel for excessive play.

3. For pools or spots of oil under transmission, universal joints, and rear end. Loosening up of flange joints or breakdown of oil seals causes oil leaks, and either the loosening or leak of oil may contribute to a road failure.

4. For pools or wet spots under the engine which indicate leaks in the radiator or crankcase. A condition that may contribute to a road failure resulting from lack of water, fuel, or oil. Similarly, water level in the radiator and oil level in the crankcase should be checked.

5. Fan belt for tension.

6. Clutch adjustment. Clutch pedal should be tried for clearance. If clutch lining wears, the position of the clutch release and subsequently the clutch pedal changes.

7. Ammeter, for any excessive draw indicates short or leak at some point in the lighting circuit.

8. Last but by no means least, the brakes, before starting the engine; when air brakes are used, by reading the air pressure gage, and by pedal pressure test for other types.

B. *Starting of engine and warmup period.* Once having assured himself that his vehicle is ready to roll, in so far as the then inert units are concerned, the driver is ready to start the engine. It seems elementary to dwell at any length on how to start the engine. Yet vehicle conservation really begins the moment rotation starts, and the failure of drivers to practice what they know contributes to unnecessary delays, undue wear, and sometimes positive and immediate, harmful results. Let's review the ABC's of starting an internal-combustion engine.

1. The transmission must be in neutral, and preferably the clutch pedal depressed, before turning on the ignition.

2. Turn on the ignition switch and press the starter switch. Be prepared to release the starter switch button immediately if the cranking motor stalls, otherwise serious damage will result. If the engine does not start promptly, wait a while before again using the starter. The starter should not be operated for more than 30 sec at a time.

3. If started cold, once the engine is running allow it to warm up thoroughly before opening the throttle under load. When an engine is cold so are its parts and oil. Cold oil cannot give adequate lubrication to engine parts. Check the oil pressure gage. At the start the gage will show a high pressure but as the engine warms up this will ease off to normal. Normal pressures vary according to the type of engine, and the driver should acquaint himself with the norm of the vehicle engine entrusted to his care. If the pressure remains high, stop the engine promptly and report the condition to the mechanical department. If the pressure drops below normal after warming up, stop the engine immediately and check for abnormally low crankcase oil level. If oil level is okay do not again start engine. but report condition promptly to mechanical department.

4. Check ammeter reading. After the engine starts, the ammeter should indicate charge as the engine speeds up. With a well-charged battery the ammeter should show a charge of only a few amps. With a completely discharged battery the charging rate goes up to the maximum output of the generator. If the driver knows that the battery is fully charged and the high charging rate continues, or if charging rate continues low after the engine is speeded up, report promptly to mechanical department.

5. Before starting the vehicle the air pressure gage (when truck is equipped with air brakes) should be built up to approximately 85 psi. The time required to build up the pressure varies, but approximately one minute may be considered representative. If required pressure cannot be built up, report promptly to mechanical department.

6. During the warmup period when not under load is an ideal time for the driver to detect untoward engine noises which are frequently the forerunner of trouble. A general engine noise is often indicative of a low oil level. Squealing may come from belt slippage, fan, generator or water pump bearings, generator brushes, or clutch throwout bearings. Light creaking or tapping may be a sticky valve, a broken valve spring, or a loose wristpin. Loud knocking is usually a burned out con rod, a broken piston, or a broken-off valve head. A dull heavy thumping is a loose or burned out main bearing. If in doubt, take no chances, but report promptly to mechanical department.

C. *Taking off.* Taking off too quickly has aptly been described as a jack rabbit start, and nothing is more harmful, all parts of the vehicle from the front axle to the differential being affected. Let's analyze briefly what happens when the energy generated by the engine is transmitted to the driving mechanism of the vehicle.

1. In order to start the vehicle the engine must have stored up a sufficient reserve of energy in the flywheel to pick up the load without materially slowing down the engine. If too much energy is stored up it is most difficult to release it through the clutch and component driving units without severe shock. If too little energy is stored up the engine will stall. Therefore, proper engine speed in starting is the initial fundamental.

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2. Once the proper engine speed is attained, it is necessary that the energy stored up in the flywheel be released with a minimum of shock. If done too slowly, excessive clutch slippage occurs and the resultant heat destroys the lining or causes rapid wear, or in some cases checks and warps the clutch plates. For best results slippage should be limited to approximately 1½ sec. If done too precipitously the adverse results may be of equal or even greater severity. A sudden engagement of the clutch results in a force greater than that for which the units were designed being applied to the transmission and all drive parts, including universal joints, ring gears and pinions, and axles. Even though these parts may not immediately fail, the strain may fracture the unit, allowing failure at a more inconvenient and expensive location.

3. It is apparent that the control of clutch slippage is the only manner in which the energy of the engine can be transmitted to the driving units without undue shock. However, the driver cannot do this with his foot alone. The selection of the proper transmission gear is also a factor in the right form of take-off. The use of the lowest transmission gear in starting is the only safe way to prevent excessive slipping. Transmission gears other than low may, under very favorable conditions, be safely used in starting, but whenever clutch slippage continues in excess of 1½ to 1¾ sec, then too high a gear is being used to start.

4. Only those mechanical units of the vehicle used as a prime mover have thus far been considered in bringing about a proper take-off. When trailer or semitrailer are used as a train the driver should allow slack in the couplings to be taken up gently before acceleration, otherwise undue strain on drawbar and pintle hook will result.

5. The starting period does not end with the initial engagement of the clutch and first turn of the wheels. It continues throughout the acceleration period until normal or desired rolling speed has been attained. Proper acceleration is as vital in conservation of equipment as any other phase of driving. Proper acceleration can be obtained only through proper use of the accelerator pedal and transmission gear shift lever.

Before the driver can accelerate to best advantage he must know the engine rpm range in which maximum torque and maximum bhp are developed and must keep the engine speed within this range through the use of the several transmission gears, progressively going up through the gears as the vehicle gains momentum. If the engine rpm falls below that point at which the maximum torque is developed, momentum is not only lost but lugging of the engine results. If engine rpm exceeds the point at which maximum bhp is attained, not only is no greater power developed but fuel is wasted and injury to the engine results. There is a variance of engine speeds at which maximum efficiency is maintained, not only from the standpoint of power but fuel consumption and long, trouble-free life as well. However, the most productive and efficient range is about 200 rpm below the governed speed, and the rpm at which two-thirds to three-fourths full torque is attained. Thus, for an engine having a governed speed of 2600 rpm, the desired speed range would be between 1700 rpm and 2400 rpm. In other words, when taking off, the minimum engine speed should be at approximately two-thirds the maximum torque or around 1700 rpm. As momentum is gained and engine rpm reaches 2400, the next higher transmission speed should be used. The engine should be kept "wound up" during the accelerating period; that is, kept as nearly as possible to the point where maximum bhp is developed.

Perhaps the most practical criterion of good acceleration is a steady and consistent "thrum" of the exhaust, punctuated by the slight kick upwards occurring at the time of shifting gears. If this "thrum" becomes erratic in volume or consistency, it is an indication that the driver is figuratively asleep at the wheel or has been improperly trained.

Driver aids, such as tachometer and vacuum gages, are invaluable contributions to proper acceleration.

2. *Don't drive too fast.*

A. As in the case of the starting operation, the admonition "don't drive too fast" has greater significance than is first apparent. Reference is made not only to the road speed of the vehicle but to the speed of the component mechanical units of the vehicle as well.

Without in any manner deprecating the ill effects of excessive road speeds, it is well to remember that serious injury to a mechanical unit can be suffered at relatively low road speeds, when the element of speed is otherwise present. Also, that great moral persuader, the highway patrolman, has no jurisdiction in matters other than road speed. Consider:

1. That high engine speeds set up high inertia and centrifugal forces, causing high main-bearing and con-rod bearing loads, and overloading other working parts of the engine. These high loads cause premature wear, overheating, and deterioration of motor oil. High engine speeds are inefficient and wasteful from the standpoint of engine and fuel economy.

2. Excessive engine speeds (above governed speed) are attainable not only when under power but when coasting down hill as well. When the engine is under power and the governed speed is approached, the next higher transmission gear should be used. If the highest gear is in use and governed engine rpm is approached, the vehicle has reached its maximum economic speed. In coasting down hill the reverse is true. The vehicle should never be permitted to roll at any speed higher than it would be driven by the maximum governed speed at the same gear ratio.

3. Once the vehicle has attained its cruising speed, excessive engine speeds may be avoided by in turn avoiding heavy gear work to the greatest possible extent. As was indicated in comments on acceleration, this may be attained by keeping the engine speed to approximately two-thirds the governed speed and by using to the utmost practical extent the truck's momentum.

B. Let's consider the other and more generally accepted phase of driving too fast; that is, the speed of the vehicle on the road. How fast is too fast? The answers are:

1. We have already indicated that if under any condition or in any transmission gear the governed speed of the engine is exceeded, the vehicle is going too fast. If in high gear the engine speed is excessive, then road speed is too.

2. If the tires run hot the vehicle is traveling too fast considering the load carried. The greater the load and speed the greater the power transmitted through the tire, and the greater will be the heat generated in the tire. Excessive heat causes premature tire failures.

3. If there is a pronounced tendency of the vehicle to roll sideways when rounding a curve to the point of a near skid, the vehicle is traveling too fast. A curve should always be approached at a speed which will not require braking and which will permit application of power.

4. If at any time there is not instant reaction to the

steering wheel, the vehicle is traveling too fast for the condition of the road bed. Washboard, gravel, and wet roads, and roads with excessive crown have a tendency to minimize steering reaction and speeds must be gaged accordingly.

5. If at any time the driver cannot bring the vehicle to a complete stop to avoid impact with an unexpected obstruction, the vehicle is traveling too fast. It should be remembered that mechanical stopping distance is not directly proportional to speed but rather is proportional to the square of the speed. Driver reaction time and down grades even though slight must be given consideration.

6. In heavy traffic any speed which necessitates frequent changes of lane is too fast for conditions under which the vehicle is traveling. Move with traffic, don't fight it.

7. If the load tends to shift on curves or upon acceleration or deceleration, speed is excessive for cargo carried.

8. Perhaps the sanest and most comprehensive admonition as to road speed is the "basic speed law" of several states which stipulates:

No person shall drive a vehicle upon a highway at a speed greater than is reasonable or prudent, having due regard for the traffic on and the surface and width of the highway, and in no event at a speed that endangers the safety of persons or property.

C. Unless the proper perspective is maintained, the foregoing seems to rule out for trucks, in the interest of conservation and safety, any speed in excess of an exaggerated crawl. No conclusion could be more erroneous. Trucks of modern design and good maintenance, coupled with proper driving and loading procedure, can be operated safely and economically at road speeds consistent with modern highway traffic flow. Were this not true the highway truck would not have attained its present eminence in the transportation field. It is only when the vehicle is pressed beyond its design capacity, or subjected to improper use, that conservation fails and hazards are generated.

3. *Don't stop too quickly.*

If at first glance there seems to be something contrary to the general scheme of things in this admonition, it is because so much stress has repeatedly and advisedly been made upon the necessity of being able to stop quickly in emergency. The interests of both conservation and safety also demand that quick stops be avoided except in emergency. This admonition also carries with it the factor of precipitous deceleration even though no stop is contemplated, for the reason that if quick stops are harmful, then all events leading up to the quick stop are in a proportional degree harmful.

All harmful effects of the jack-rabbit start are present in the cowboy stop. The entire mechanism of the vehicle is similarly and adversely affected. We are all familiar with the treatment accorded automobiles in public parking lots particularly with respect to the senseless and destructive practice of stopping on a dime. We have watched with dismay a car driven at excessive speed into a relatively small parking space, and at the last moment the full force of the brakes viciously applied. The immediate screaming of the tires and the sickening downward sag of the springs give audible and visual evidence of the torture to which the vehicle as a whole has been needlessly subjected. Apply this same treatment to a truck and the ill effects are the same, but in a much greater degree because of the increased weight and increased momentum of the latter.

There seems no purpose in dwelling at any great length on the evils of a cowboy stop as none but the most inexperienced and possibly vicious truck drivers indulge in the practice. However, the second cousin of the cowboy stop, that of improper or too precipitous deceleration, is too frequently present, and remedy can be had only through better education in the use of brakes. Let's review good braking procedure.

A. The only manner in which the movement of a vehicle is retarded is through the use of friction between the tire and the road bed. Brake drums, brake lining, brake shoes, and the several means of actuating the brakes properly are but a means to this end. The amount of traction between the tires and the road surface is dependent upon the coefficient of friction between the tires and the road surface. It is, therefore, important that in any brake application the driver take into consideration the nature of the roadway surface and know just what reasonably to expect. The relative tractive effect of varying road conditions ranges from approximately 20% on ice and increases proportionately upward for packed snow, gravel, wet concrete, and dry concrete, approximating 90% on the latter, speed, of course, being an influencing factor.

B. Notwithstanding the fact that brake lining, shoes, and the actuating mechanism are but means to an end, they are a most important and vital means. If they are faulty or inoperative there will be inadequate retarding effect on the tire, and regardless of how favorable the road condition, momentum will not be adequately checked. Although the driver is not generally responsible for maintenance of mechanical units, it is important that he recognize and report conditions that may cause either immediate trouble or build up unnecessary expense, such as:

1. When brake pedal goes to the floor board it means that the brake shoes are not properly adjusted, leaks in system, wear in system, pedal improperly set, or no fluid in the supply tank.

2. If all brakes drag it may mean that there is mineral oil in the system or the pedal is improperly set.

3. If one wheel only drags it may indicate a broken brake shoe return spring, clogged or cramped hydraulic brake lining, brake shoe bar seized to bearing pin, brake shoe set too close to the drum, piston cups distorted, or loose wheel bearings.

4. If the truck pulls to one side possibly the lining is grease soaked, shoes may be improperly set, braking plate loose on rear axle, tires not properly inflated, or clogged or cramped hydraulic line.

5. If pedal is springy or spongy there may be wear in the system or the brake shoes may be improperly adjusted.

6. If the brakes are overly sensitive, brake shoes may not be properly adjusted or there may be wear in the system.

Any of these conditions are an impediment to proper deceleration and if permitted to continue may cause immediate hazard and excessive future expense.

C. Good brake equipment is of no avail if not used properly. The fundamentals of correct usage are simple and understandable, that is:

Under normal driving conditions:

1. Anticipate slowdown or stop.

2. Avoid harsh, severe applications.

3. So control the vehicle stop that at the end of the application the brakes are almost released.

When descending grades:

1. Leave truck in gear and, with accelerator released, make use of engine compression.

2. When so using the transmission gear, never permit engine rpm to exceed the governed engine speed.

3. On extended grades make intermittent brake applications to minimize tendency of drums and shoes to overheat.

4. Never let the vehicle even momentarily get out of control.

Use of parking brake:

The hand (or parking) brake is generally of a propeller-shaft type and as such has never warranted the name of an emergency brake. Its usage should be restricted to parking. Regular use otherwise will not only result in short life, but extreme heat may generate with high road speeds, which may so weaken drum and disc as to make them burst.

4. *Don't continue to drive when there is some apparent defect, mechanical or otherwise, in the vehicle.*

In any discussion of this admonition we must necessarily presuppose some knowledge on the part of the driver of the mechanical functioning of his vehicle. The minimum requirement should be that he understands fundamentals, such as what an engine is and the basic functioning of the clutch, transmission, universal joints, drive line, steering mechanism, differential, axles, and brakes. No technical knowledge is necessary, nor should repair ability other than that coming under the heading of "driver care" be required Any conscious or unconscious invasion by the driver into the mechanical field, other than in emergencies, is preponderantly more harmful than beneficial, not merely because of possible lack of mechanical skill on the part of the driver, but principally because of the divided responsibility thus brought about.

Knowing fundamentals, the driver can and should recognize danger signals when they occur, and once warned, should, if circumstances warrant, immediately cease operation of the vehicle and report this condition to a superior. The extent to which operation may be continued, even though difficulty is experienced, is and must be dependent upon the training and good judgment of the driver.

Outstanding danger signals indicating immediate stoppage of operations, or at least warranting immediate further investigation, are (a) unusual engine noises (see previous comments on starting and warmup period); (b) excessive engine temperature; (c) loss of oil pressure; (d) loss of air pressure or brake pedal pressure; (e) erratic steering; (f) dragging brakes; (g) unusual noises or vibrations.

Danger signals which should be promptly reported at the end of the trip are:

(a) Any evidence of leak in cooling system, engine transmission or rear end; (b) hard starting, erratic ammeter, high exhaust back pressure; (c) sluggish acceleration; (d) excessive play in steering wheel, hard steering, shimmy, dive, tramp, or wander; (e) excessive clutch slippage or retarded clutch action; (f) dim or faulty lights; (g) unequalized brakes; (h) tendency of engine to overheat; (i) any unusual noises in the vehicle otherwise.

5. *Don't hit anything.*

This admonition seems so elementary as to reflect on the intelligence. Viewed in the generally accepted sense, this is perhaps true. In any event, avoidance of collision and impacts of a similar type more properly belong in a discourse on operating safety. However, there is another phase of collisions that contribute greatly to truck deterioration. Reference is made to those minor impacts not sufficient within themselves to warrant great concern, but which through repetition start in motion forces which often have serious and costly consequences, such as:

A. In parking, avoid scraping or bumping of the curb. Not only is tread and side wall damage suffered, but wheels and axles are thrown out of alignment, resulting in erratic steering and rapid tire wear.

B. Avoid hitting minor obstructions on streets and highways, such as traffic buttons, holes or miscellaneous loose material. The impact frequently damages the carcass of the tire, resulting in premature failure or even blowout.

C. Avoidance of impact applies not only to the vehicle proper but the load on the vehicle as well, and any equipment which may be carried. Excessive load projections may side-swipe passing vehicles or objects. Improperly secured equipment, such as barrel skids and ladders, may work outward and foul passing objects.

6. *Don't let anything hit you.*

In this one admonition the realm of operating safety is advisedly invaded. The only manner in which one can avoid being hit is to keep out of the way of others. And assuming the vehicle to be in representative mechanical condition, the only way in which one can keep out of the way of others is to practice what the Atlantic Refining Co. so aptly calls "defensive driving" in their booklet on motor-vehicle driving practices. A "defensive driver," says the Atlantic Refining Co., is one who makes allowances for the lack of skill and lack of knowledge on the part of the other fellow – who recognizes that he has no control over the unpredictable actions of other drivers and pedestrians – and who therefore develops a defense against all these hazards. He concedes his right of way and makes other concessions to avoid collision. He is careful to commit no errors himself and is defensively alert to avoid hazards created by weather, roads, pedestrians, and other drivers.

7. *Don't load improperly or excessively.*

Good driving practice and good maintenance practice can be offset readily by improper or excessive loading. Improper loading may occur even though the resultant gross vehicle weight is well within the limitations of the vehicle. A vehicle may be considered improperly loaded if (a) the load is insecurely fastened or is liable to shifting or loss; (b) the load is not properly distributed either as to front and rear axles or as to the right or left side of the vehicle.

Excessive loading is present when the rated capacity of the vehicle is exceeded, or if, regardless of rate of vehicle capacity, the load results in undue strain on the vehicle due to road conditions.

Improper loading generally results in undue strain being placed on one axle or one section of the vehicle. Excessive loading results in undue strain on the entire vehicle. Ill effects are generally experienced as follows:

A. Excessive axle loads make excessive tire loads.

B. Excessive axle loads may result in misalignment due to housing being permanently sprung or temporarily deflected while under load.

C. Excessive loads may result in sagging or broken frames, and spring distortion and breakage.

D. Excessive loads when accompanied by undue speed are one of the causes of short wheel-bearing life.

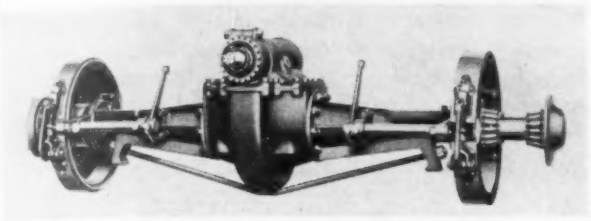
E. Excessive loads consistently carried necessitate undue gear work, and heavy starting shortens the life of the clutch and transmission.

F. Improper loading, in addition to placing an undue strain on some particular portion of the vehicle, may affect proper steering due to improper weight distribution.

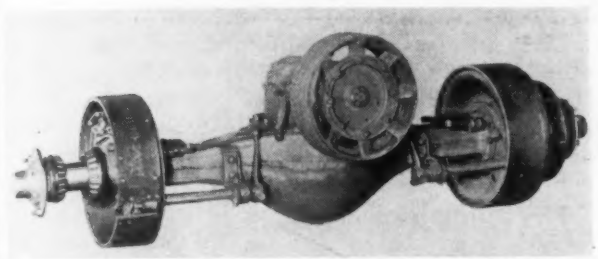
BRAKE DESIGNS and Methods of RATING BRAKES

For Commercial Automotive Vehicles

by RALPH K. SUPER
Brake Division Engineer
Timken-Detroit Axle Co.



▪ Fig. 1 – External bands used on brake drum



▪ Fig. 2 – Internal expanding brake of band type

VIEWING in retrospect the designs of brakes for commercial automotive vehicles, it is not difficult to establish the fact that many of the current problems have existed from the beginning. Some of these problems have developed as the result of manmade limitations, but many are basic problems in the application of physics and mechanics. Vehicle brakes have always had the job of converting the energy of motion into heat energy and then dissipating it. In early brake designs, the size of the brake was influenced largely by the holding or stopping ability of the brake and not by any requirement of capacity for heat dissipation. Solid rubber tires and wood wheel construction provided a combination which ensured adequate space for the brake mechanism. In fact, enough room was available to permit the use of external bands on the drums (Fig. 1). Brake torque ability persisted as the major problem and power braking had not been made available for automotive use. The manual effort of the vehicle operator was transmitted to the brakes through mechanical links and shafts. Brake drums were cast steel or steel stampings, brake lining was woven, and toggles were quite generally used to actuate the brake.

In an effort to improve the power of the brake and to permit larger diameters, the internal expanding brake was adopted. This was a band brake (Fig. 2) or better still a shoe brake (Fig. 3). A cam was introduced into the latter design. It was currently called a "square" or "box type" cam.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 7, 1946.]

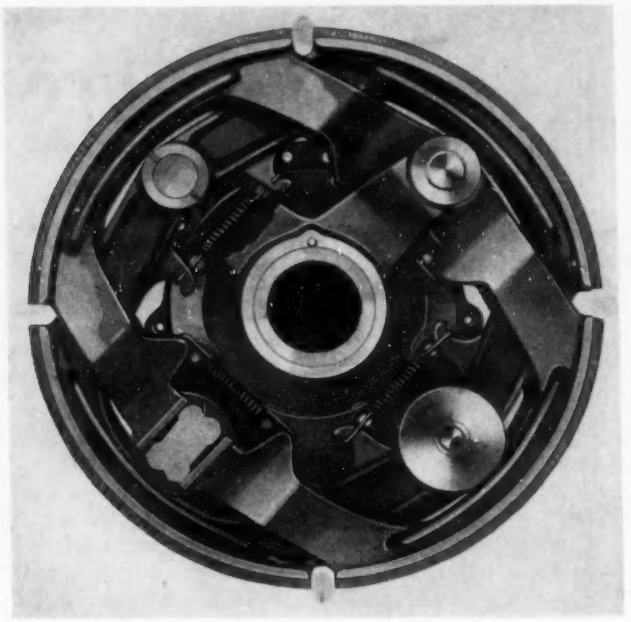
Although burdened with these problems, the solid tire equipment acted as a limiting factor on vehicle speed. The introduction of pneumatic tires on commercial vehicles created many of the current difficulties. Speeds were increased quickly and brakes on all wheels became a necessity to get both heat dissipating capacity and stopping ability. The inequalities of distribution of the driver's "foot power" to the several brakes on the vehicle finally resulted in the introduction of the hydraulic brake (Fig. 4). Power actuation supplemented or replaced manual effort and brake designs rapidly changed from a light construction (Fig. 5) to a heavy construction (Fig. 6). Brake drums changed from steel to cast iron, and brake linings were molded to secure maximum density and greater resistance to heat and wear. Innovations in cam designs and power unit mountings quickly followed (Fig. 7).

OF the various methods of establishing the size of vehicle brakes that have been in use the author points out that the ones found most satisfactory have been based on the fundamental fact that the brake is an energy-converting unit.

Formulation of rating factors at the present time, he says further, shows promise of incorporating the effect of engine horsepower on the brake factors by introducing a time element.

For those advocates of a simple method of rating brakes on the basis of liner area, Mr. Super recommends that this area be established on the basis of the projected length of the liner so that the most efficient use of the liner material is obtained.

THE AUTHOR: RALPH K. SUPER (M '27) has been specializing in brake engineering since his graduation from the University of Michigan in 1925. Currently in the Brake Division of Timken-Detroit Axle Co., where he has been employed since 1939, Mr. Super was previously associated with the Mack Truck Co. and General Motors Proving Ground.



■ Fig. 3 – Internal expanding brake of shoe type

Although substantial progress is evident, the specter of spacial limitations became more important in the design of motor vehicles. Limitations on overall width of the vehicle and the desire to limit the outside diameter of the tires had a direct effect on the space available for installation of wheel brakes. The present design of heavy-duty rear axle with dual low-pressure tire equipment indicates the very compact arrangement of the brakes (Fig. 8).

Regardless of dimensional limitations, there has persisted throughout this evolutionary period the feeling that large diameter brakes and maximum lining coverage ensured the best stopping ability and maximum lining life. It has been a slow process, but not too difficult, to establish a maximum diameter of brake. It was found to bear a direct relation to the diameter of the tire rim. Limiting temperatures of the tire bead quickly established the spacing between the drum and tire rim for the various types of operations. On this basis, larger diameter brakes require correspondingly larger base tires or the resultant adverse effect on tire life is quickly evident.

It has not been as easy to establish the facts relating to the mileage life and work capacity of a brake. Maximum lining area unfortunately continues to be the index of high capacity and good mileage. Current designs testify to the prevalence of this fact (Fig. 9).

With this background the problem of establishing a brake specification for a vehicle does not appear involved. The torque delivery of a brake can be readily established or measured in the laboratory or in the field. This factor is not the subject of discussion of this paper. The specification indicative of mileage life and work capacity of a brake is the subject to be reviewed and assumes expanding importance with every increase in speed and loads.

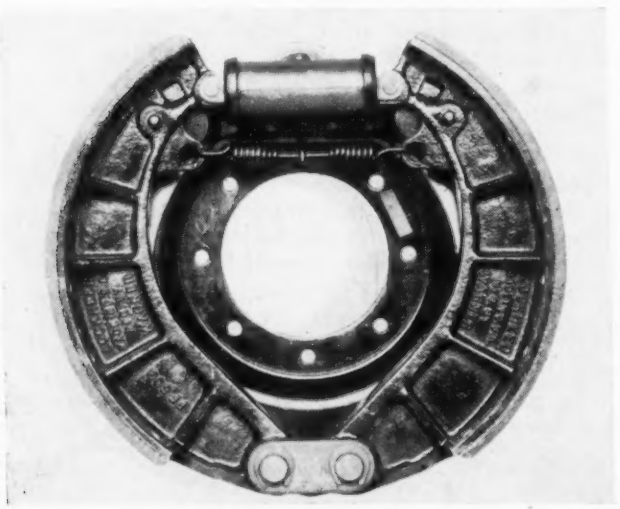
■ How Big Should Brakes Be?

How big should the brakes be on a truck and how big should they be on a bus? How much lining area should the brakes have? Does this lining area ensure heat dissipating capacity in the brake and satisfactory mileage life?

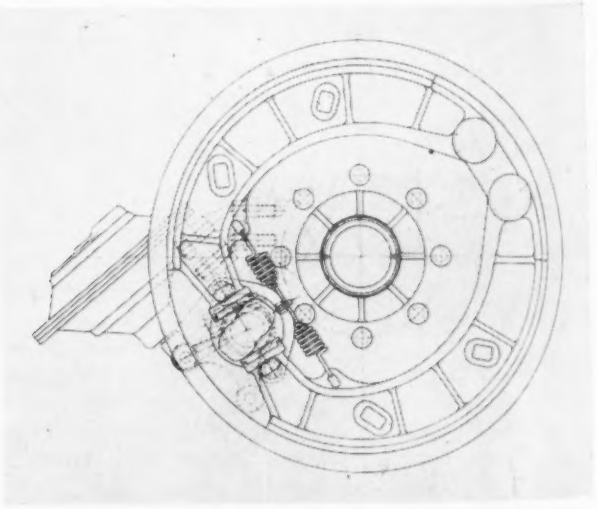
These questions are typical of those presented regularly to the brake engineer by the operator and vehicle manufacturer. There have been several methods of analysis for determining brake sizes which have been quite widely used. Figures based on these methods are frequently incorporated in vehicle specifications.

The purpose of this paper is to review the current methods of analysis and indicate particularly where they are capable of improper interpretation and misuse. Corrective suggestions based on such information as is available will also be included. It is believed that a summation of this information will stimulate thinking along the line of formulating a standard and universal method for determining the size and capacity of automotive brake equipment.

Reference to the chassis specifications in any of the automotive magazines will reveal that the wheel brake sizes are listed in the tabulation and also, the corresponding lining areas. The availability of the lining area infor-



■ Fig. 4 – Hydraulic brake



■ Fig. 5 – Brake of light construction

mation has made it a simple matter to use this figure as a basis for determining the adequacy of the vehicle brakes. Over a period of 15 years it has been determined from field experience that a truck having a gross vehicle load and lining area ratio so that a figure of 40 lb gvw per sq in. of lining prevailed would give satisfactory brake performance under average conditions of operation. Experience alone indicated whether the figure 40 should be increased or decreased to meet variations of these conditions. For instance, brakes used in transit bus service are given a rating of 25 lb gvw per sq in. of brake lining based on the seated load, due to the greater frequency of stopping.

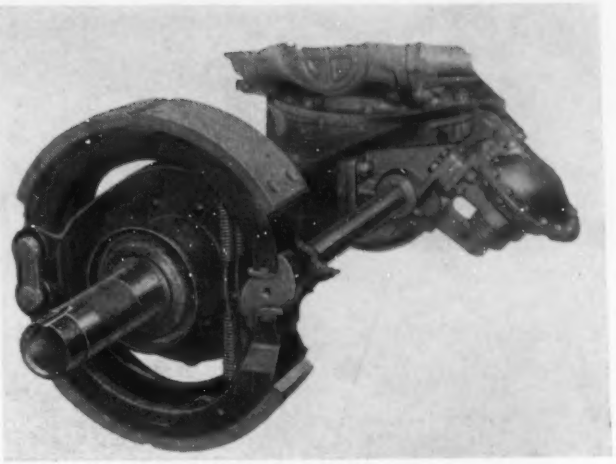
There is no question about the convenience of dividing the vehicle weight by 40 or 25 and checking the resulting figure obtained against the total brake lining area. The accuracy of this method is questionable, however. As an example, it is a known fact that the overall width of the vehicle is restricted and that the space available for brakes is limited. This is particularly true where dual tires are mounted, such as on the rear axle. It is not uncommon for the area-weight ratio for the brakes on the rear axle to be higher than desired or specified. As a consequence, the brakes on the front axle are oversized and a low factor obtained for that location. When the figures for both front and rear axles are combined a satisfactory overall factor is indicated. The fallacy of this reasoning is that it is impossible, due to steering difficulties, for the front brakes to handle the proportion of the vehicle load indicated by their area, and consequently, the rear brakes must work under an overloaded condition with subsequent unsatisfactory results (Fig. 10).

The simple answer to this problem is that the lining area figures should be separated into those existing on each axle. The load figures thus obtained will be more nearly correct.

Effect of Increasing Lining Area

In utilizing the area of the brake lining as the basis for determining the adequacy of the vehicle brake equipment, it is only natural to assume that the greater the lining area the more the load capacity of the brake. The tendency, therefore, is to add lining to the brake shoes until the entire braking surface of the drum is covered. On some brakes, particularly hydraulic brakes utilizing so-called long and short linings (Fig. 11), it is possible to increase the lining area on one shoe approximately 30%. It is seldom questioned whether the addition of this area of friction material contributes any improvement in the performance of the brake, even though the load area factor is better. The fallacy of this reasoning is evident. Adding lining on a brake shoe to obtain greater area and capacity is not the answer. The real evidence of the error of this reasoning is the amount of lining still remaining on the brake shoe at the time relining becomes necessary. This lining cannot be used but must be thrown away.

It is not desired to enter into any complicated mathematical analysis of this conclusion, but a few simple sketches may indicate the reason for the actual results that are obtained. Assume that a brake shoe has an arc length of 180 deg, as shown in Fig. 12, and that the lining extends over its full length. The shaded segment indicates that portion of the lining which will be worn from the shoe if the shoe is moved along a radius bisecting the shoe



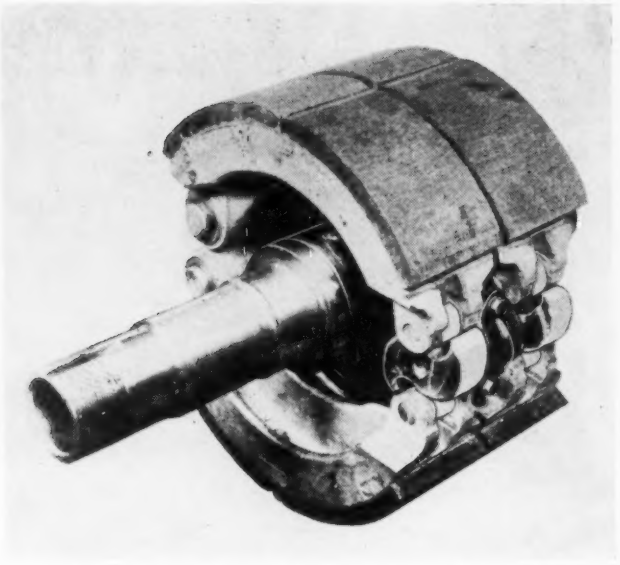
■ Fig. 6 – Brake of heavy construction

arc. The two checkered segments indicate the lining remaining on the shoe. Reference to Fig. 13 will show the 180-deg arc divided into 30-deg segments. The shaded area is shown grouped into three arc lengths, namely, 60 – 120 – 180 deg. If a planimeter is run over each of these areas, it will be found that the shaded area within the 60-deg sector constitutes 50% of the entire shaded area. Secondly, the shaded area within the 120-deg sector constitutes 86.6% of the total. The remaining 13.4% of the area is within the 180-deg sector.

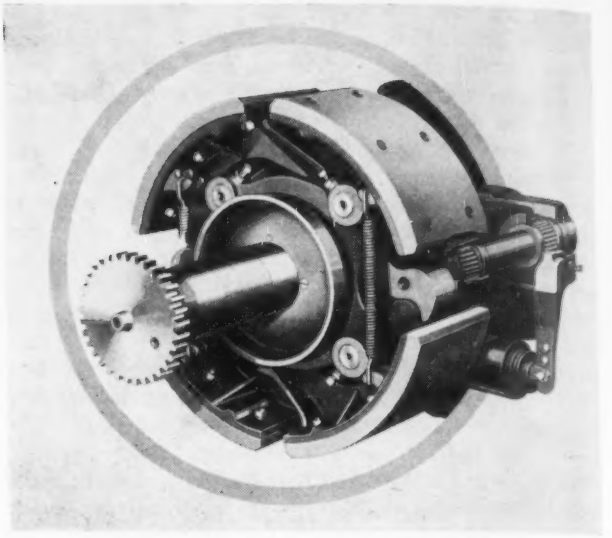
These percentages are significant in that they indicate that one-half of the brake lining contributing to the life performance of a brake is in a zone of 60-deg length at the center of motion of the brake shoe. Also, 86.6% is in the 120-deg zone shown. The center zone, therefore, is one-third of the total length and provides 50% of the lining material. The 120-deg sector is two-thirds of the total arc length and provides 86.6% of the material. The two end segments constitute one-third of the total arc length and provide the remaining 13.4%. The material in the end segments is only 26% as valuable as the material in the center sector. The cost of the material is the same for either location. Note, too, that the major part of the lining remaining on the brake shoes (checkered area) is in the end segments and will be discarded.

It should not be difficult to determine the limitations of

■ Fig. 7 – Brake with power unit mounting



■ Fig. 8 – Compact arrangement of present-day brake



■ Fig. 9 – Brake with maximum lining area

lining length on the shoes based on the above percentages. It becomes a question of economics as well as engineering. These figures further indicate the value of correct positioning of the lining on a brake shoe to secure its most efficient use. It would be extremely valuable and desirable if the vehicle load lining area relation could be established so that it would automatically minimize the value of lining which could not be used efficiently for the improvement of the brake. Reference to the percentages shown for the shaded zones in Fig. 13 discloses that in the 60-deg sector, which extends 30 deg on each side of the radius through the center of movement of the shoe, a value of 50% is indicated. This percentage is exactly the same as the sine of half the sector angle. Half the sector angle is 30 deg with a sine function of 0.5 (Fig. 14). These sine functions when scaled along a diameter at right angles to the direction of the shoe movement constitute the chord of the arcs projected onto the diameter.

The center sector of 60 deg projects a length equal to 0.5 of the diameter, while the 120-deg sector projects a length of 0.866 of the diameter. The remaining end segments cover 0.134 of the diameter.

The translation of this concept of using the projected length of the brake lining as a basis for determining lining area would mean that a brake would have to be analyzed by drawing a diameter through the shoe anchor pin and a radius established at right angles to this diameter. This radius would indicate the line of motion of the shoe (Fig. 15). The lining arc would then be referenced onto the diameter by lines parallel to the radius (Fig. 16).

■ Projected Area of Lining

The factor of 40 lb gvw per sq in. of lining as used for trucks was based on a lining length on the shoe equivalent to the arcuate length of the segment. This factor must be increased if the projected length of the lining is to be used rather than the arcuate length as a basis for establishing the lining area. The relation of the arcuate length to a full diameter is 1.57. This figure would change the 40-lb factor to approximately 63 lb. Although this appears to be correct, it is known that most conventional two-shoe brakes have a lining arc length of 110-130 deg, so that a ratio between the arcuate length of a 120-deg arc and its projected length would be more nearly representative of a practical load-area figure based on current designs. The ratio of the lengths in this range is about 1.21 to 1. This figure then would increase the 40-lb factor to 48.4, or a round figure of 50 lb gvw per sq in. of *projected* area of brake lining. The figure for bus brake rating would be 30-32 lb weight per sq in. of *projected* area of the lining.

The use of *projected* areas instead of arcuate areas as a means of rating brakes does not present difficulties to either the engineer or the operator either in securing the information or interpreting the results. The direct relation between the lining material on the brake shoe, which may be removed by wear, and the *projected* area of this material justifies the use of this method of rating brakes on an area-load basis.

■ Actual Pressure on Lining

A second method of rating brakes also involving the use of lining areas is one in which the unit *pressure* on the lining, as exerted by the actuating mechanism, is used as a basis for performance. A figure of 150-200 lb pressure per sq in. of lining is used for this rating and the area involved is that based on the arcuate length of the liner. This method is not widely used because of the necessity for having complete information on the mechanism for developing the forces on the brake shoes. The method is mentioned here because it will further indicate the diversity of methods of rating brakes. This method also encourages the addition of lining area with no consideration of its usefulness.

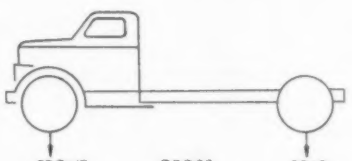
A rigid brake shoe being forced against a brake drum, as shown in Fig. 15, is similar in many respects to a shaft bearing in a journal (Fig. 17). Reference to any handbook of mechanics reveals that the load on the journal is vertical and varies from a maximum at the center to zero

at the sides. Also, the area supporting the load is that obtained by considering the diameter of the shaft and the length of the journal. This area in a journal is called the *projected* area. The application of this principle to a brake shoe would again mean that projected lining areas would be more accurate than arcuate lining areas in establishing the unit pressure loadings based on the above method of rating brakes. The same correction factor as used in the vehicle load-lining area method would also apply here. This was 1.21 and the figure for unit pressure on the brake lining would be increased to 180-240 lb per sq in. of *projected* area of the lining.

It is hoped that this brief review of the load-area and pressure-area concept of rating brakes has indicated the necessity for analyzing the brake capacity of each axle as a separate unit and further that the *projected* area of the lining is a more accurate basis of rating than the arcuate area factor now in use.

The paper thus far has limited the analysis to conventional two-shoe brakes. There are also brakes currently in use that have more than two shoes and that have a lining coverage approximating the entire braking surface of the brake drum. The mechanics of the design of these brakes ensures the economical use of the lining material and the throw-away portion is a minimum. Further, the lining has not been added with any idea of securing an unwarranted advantage. These brakes stand out in the tabulation of lining areas, as size for size they have almost 50% greater lining area. In the application of the load area factors previously mentioned, these brakes have 50% greater capacity than conventional two-shoe designs.

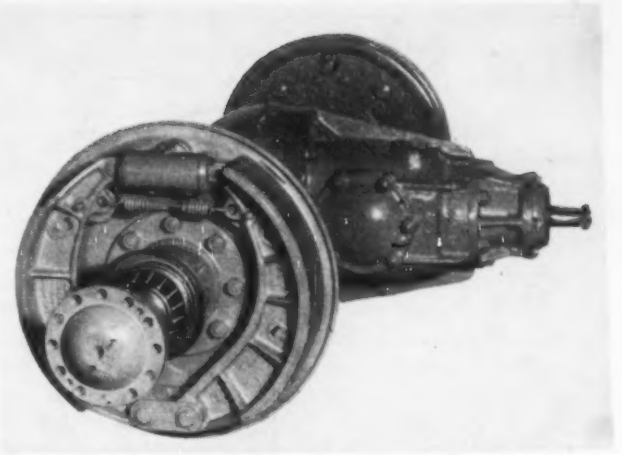
These brakes emphasize the extreme variations in capacity which are possible using the load-area method of rating brakes. The capacity differences show a disregard for the basic principle of the utilization of a friction device for converting the energy of motion into heat energy. The brakes as used on commercial vehicle axles are such a device and the heat energy generated during the braking action must be dissipated. In most cases the air in contact with the brake drum is the medium for carrying off the heat and, assuming that this area is a function of



	FRONT	GROSS	REAR
VEHICLE WEIGHT-GROSS		15000	
LINING AREA—TOTAL		334	
G.V.W. PER SQ. IN. LINING		45	
WEIGHT DISTRIBUTION (LBS.)	3600		11400
WEIGHT DISTRIBUTION (PER CENT)	24		76
LINING DISTRIBUTION (SQ. IN.)	128		206
LINING DISTRIBUTION (PER CENT)	38		62
WEIGHT PER SQ. IN. LINING (EACH AXLE)	28		55

AVERAGE SPECIFICATIONS ON (4) VEHICLES

■ Fig. 10 – Distribution of vehicle weight versus brake lining area



■ Fig. 11 – Brake with long and short linings

the diameter and width of the brake, it is axiomatic that all brakes of the same size have the same capacity.

■ Work Formulas

It is on this principle of the brake as an energy-converting unit that so-called "work formulas" for rating brakes have been evolved. Such a formula would include an expression indicating the energy to be absorbed and also one showing the size and effectiveness of the brake. The work factor would have to be interpreted on the basis of experience with the type and severity of the service to which the brake is subjected. A formula of this type which has been in use for many years with very good results is known in the trade as the Timken work formula. It is briefly as follows:

Work energy = Weight on wheel (lb) × Rolling radius of tire (in.) = $W \times R$

Brake capacity = Brake width (in.) × Brake diameter (in.) × Effectiveness (brake diameter in in.) = $W \times D \times D$

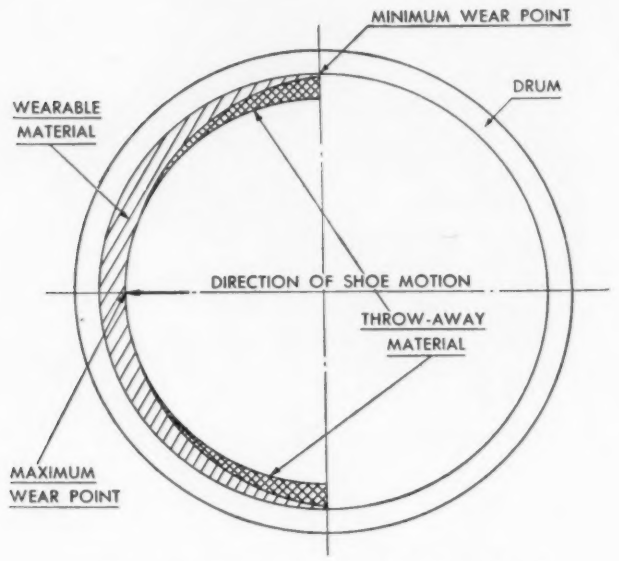
therefore

$$\text{Work factor} = \frac{\text{Work energy}}{\text{Brake capacity}} = \frac{WR}{WD^2}$$

A work factor of 110-115 has been found satisfactory for trucks. A work factor of 80-85 has been successful for buses. Approximately the same differential exists between these factors as prevailed between the load-area factors of 40 and 25 used under the original methods of load-area and pressure-area rating. The above work formula can be criticized from many angles, but the fact remains that it contains the essential expressions for work and capacity and that it is simple, useful, and has proved itself.

It will be noted that no lining area factor is introduced, but merely the diameter and width of brake.

A further consideration of the results of brake operation in actual service will indicate that the analysis of brake equipment based on the heat capacity of the unit



■ Fig. 12 – Lining wear curve for 180-deg arc length

is the most logical. For instance, the ability of a brake drum to dissipate heat is increased either by permitting more drum area to contact the surrounding air effectively, or by causing more air to circulate over the drum surface. The former is accomplished by adding cooling ribs to the brake drum, while the latter is obtained by coring ventilating holes in the brake drum dish. Adequate clearance between the tire rim and outer periphery of the brake drum is also helpful. In heavy-duty brake service, in recent years, practically all wheel brakes are designed so that the shoes are mounted on the open type of spiders. No dust shields or covers are necessary with this construction and, as a consequence, air can circulate into the inside of the brake drum. This is particularly beneficial in hot summer weather. Operators can provide testimonials regarding the improvement in brake mileage and performance due to the use of ventilated drums and the open type of brakes.

The object in introducing these facts at this point is that they indicate a feature of brake design which is extremely important and which can best be illustrated as follows:

Test figures show average temperatures for a brake drum under severe service conditions to be 400 F on the outside of the brake ring for a comparable temperature of 1100 F on the inside braking surface. The surrounding air temperature generally averages 100 F. Generalizing somewhat, it is reasonable to say that the heat dissipating capacity of the outside of the brake drum approximates the product of the circumference of the drum, the width of brake, a combined heat dissipation constant, and the differential in temperature between the drum surface and outside air. With the temperatures mentioned, the formula for drum heat dissipating capacity would be:

Heat dissipating capacity – outside (approximate)

$$= \pi \times \text{Diameter} \times \text{Width} \times \text{Constant} \times (400 - 100\ \text{F})$$
$$= 300 \times \pi \times D \times W \times C$$

Results from the field justify the assumption that the inside area of the brake drum can also contribute to the total heat dissipating capacity of the brake drum. For the purpose of this example, it is convenient to use the inside diameter of the brake drum as 0.9 of the outside diameter so that the dissipating capacity of the inside surface would then be:

Heat dissipating capacity – inside (approximate)

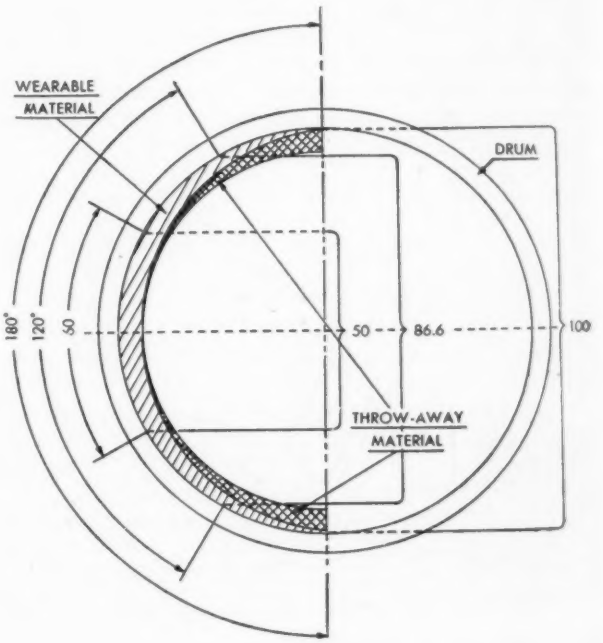
$$= \frac{360\ \text{deg} - \text{Total lining arc length}}{360\ \text{deg}} \times \pi \times 0.9 \times D \times W \times C \times (1100 - 100\ \text{F})$$

If the lining covers the entire braking surface of the drum, the first factor will be zero and no additional capacity will be obtained from this surface by circulating air in the brake cavity. If, however, brake shoes with a total lining coverage of 240 deg are used as a basis for capacity, the following figures are obtained:

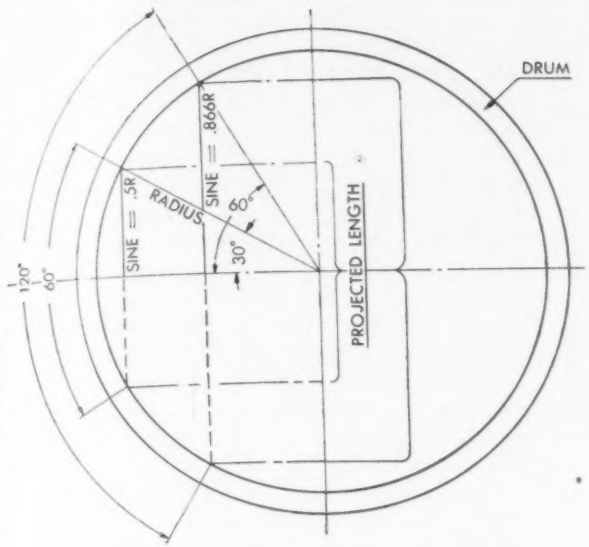
Heat dissipating capacity – inside (approximate)

$$= \frac{(360 - 240)}{360} \times \pi \times 0.9 \times D \times W \times C \times 1100 - 100\ \text{F})$$
$$= 0.3 \times \pi \times D \times W \times C \times 1000$$
$$= 300 \times \pi \times D \times W \times C$$

Combining the exposed inside and outside surfaces, the total heat dissipating capacity of the brake drum can thus be doubled by removing one-third of the lining and exposing the braking surface



■ Fig. 13 – Wearable lining material relation for equivalent arcuate segments



■ Fig. 14 – Relation of chordal and projected lengths

Heat dissipating capacity – combined inside and outside (approximate)
= Heat dissipating capacity (inside) + Heat dissipating capacity (outside) = 300 ($\pi \times D \times W \times C$) + 300 ($\pi \times D \times W \times C$) = 600 ($\pi \times D \times W \times C$)

The high temperature of the inside surface provides the differential with the circulating air. Such heat as is lost from this surface need not be carried through the drum. This is not a theoretical analysis which has no factual basis in actual vehicle operation. The results obtained in the field have also been checked in the laboratory. The following discussion will elaborate on this point:

■ Laboratory Test

In a recent laboratory test, a conventional, two-shoe fixed-anchor air-operated heavy-duty bus brake was installed on the dynamometer. The brake was subjected to a series of stops simulating transit bus service. The torque output of the brake was held constant to secure a retarding rate equivalent to 10 ft per sec per sec. The cam torque input was varied to obtain this figure. The object of this test was to determine the effect on the performance of the brake of decreasing the length of the brake liners on the two brake shoes. The original shoes had liners 116 deg long. Duplicate tests of 500 stops each were made on each set of liners, which were reduced in length by removing 5 deg of arc length of the liner from both the cam and anchor end. The changes in length in 10-deg increments per shoe were continued until only 56 deg of liner material was left on each shoe. Observations of drum temperatures and operating pressures were made and recorded during the test and the thickness of the liner was measured at the conclusion of 500 stops.

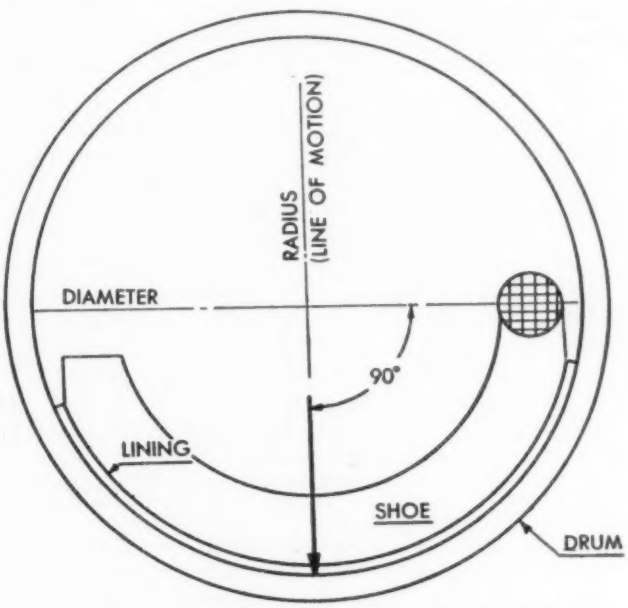
The results of this test are shown in Fig. 18 in the form of curves. It is important to note that with the decrease in liner length the drum temperature decreased, with resultant reduction in the rate of liner wear. Also, it is interesting to note that the air pressure required to make the stop shows a decrease. This fact indicates that the brake is more effective. The curves reach a minimum point at 96 deg of liner length per shoe, after which an increase in the various factors is evident.

This is only one of a number of tests and observations which substantiate the premise that the brake is an energy-converting unit and that its capacity is dependent on its ability to dissipate the heat energy into the surrounding air. It further indicates that a properly ventilated brake is penalized by having too much lining coverage on the drum braking surface.

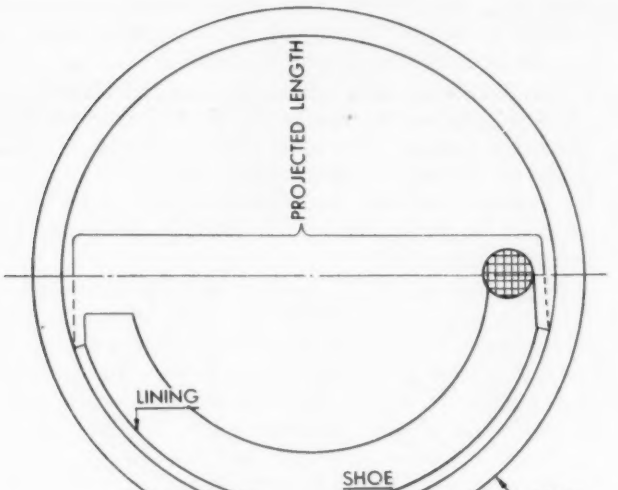
Several designs of brakes in current use in other applications than automotive wheel brakes utilize these principles of maximum air circulation and maximum exposure of the drum braking surface. Two of these are propeller shaft brakes, one a drum type and the other a disc type. A disc design of brake with minimum drum coverage is also used for high-speed train braking and an aircraft wheel brake is quite radical in the utilization of this fundamental principle. Several small discs of lining constitute the friction material on this brake.

Summarizing the points thus far presented, two methods of rating brakes on the basis of lining areas are in current usage: first, capacity based on pounds gross vehicle weight per square inch of lining and second, pounds of brake applying force per square inch of lining area. An analysis has been offered to indicate the greater accuracy of using the *projected* area of the lining rather than the arcuate area.

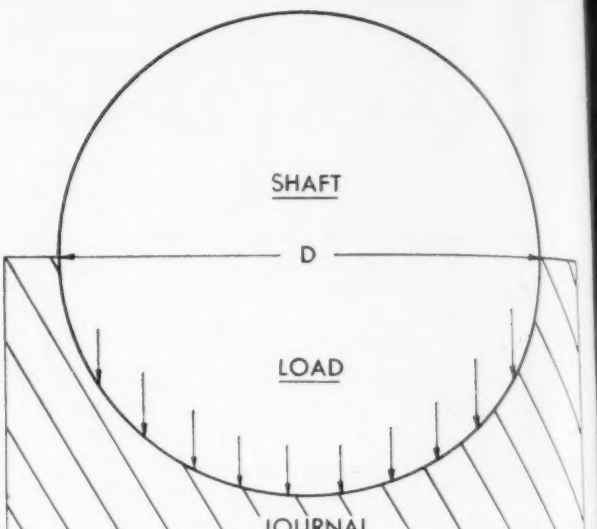
The rating of brakes based on a capacity or work factor has also been presented along with the facts relating to the effect of drum radiating surface on the capacity of the brake. When this method has been used and the factors properly interpreted, it has given the best results of any of the rating methods. It does not tend to encourage the use of an excessive area of brake liner and correction factors can be introduced to compensate for features of



■ Fig. 15 – Line of motion (instantaneous) of brake shoe



■ Fig. 16 – Projected length of brake lining



■ Fig. 17 – Direction of load of shaft in journal

design which have been found in actual service to influence the capacity of the brake.

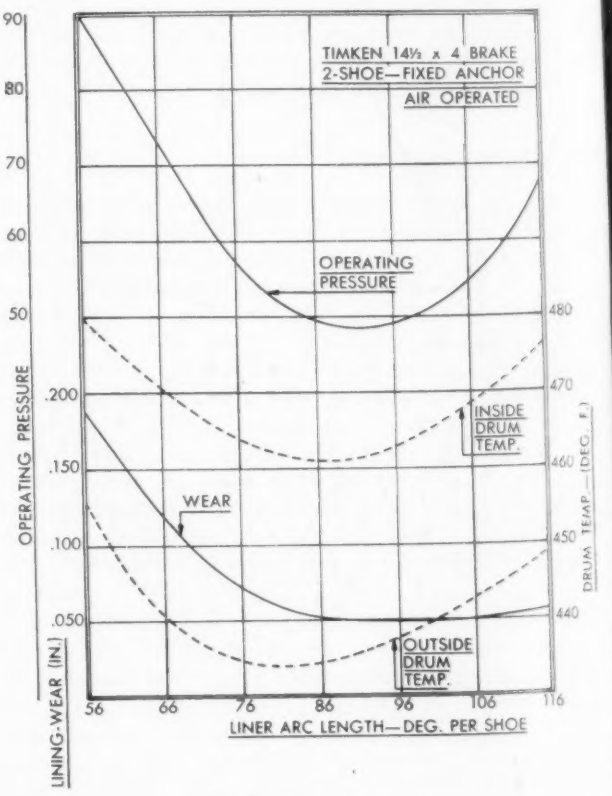
■ Effect of Horsepower

The one factor which is not prevalent in any of the formulas is that indicating the influence of the horsepower of the engine in the vehicle on the brake requirements. The work that the brakes must perform is dependent in many cases on the ability of the engine to accelerate the vehicle. This is particularly true of the transit type of bus. In this type of service, the brake requirements are almost directly proportional to the engine horsepower. For other types of vehicles the top speed is the influencing factor in brake design. This top speed also has a horsepower requirement. The manner in which a correction factor for horsepower can be introduced into the methods of rating brakes discussed so far has not been established at this time.

Much work has been done with the object of including engine horsepower or potential vehicle horsepower in the analysis of the brakes. In this approach to the problem of establishing brake capacity, the automotive brake is also assumed to be an energy-converting unit from which it is necessary to dissipate the heat energy to the surrounding air. The brake drum surface or area serves as the basis for this analysis. The rating factor is the kinetic energy absorption per square inch of brake drum swept area. This swept area is the product of the inside periphery of the brake drum and the width of the brake shoe. On aircraft brakes a kinetic energy of 12,000 to 15,000 ft-lb of energy per sq in. of drum surface is used. The application of these factors to automotive brakes is not quite as simple for the following reasons:

In the aviation field, the maximum braking speed and rate of deceleration can be established approximately from the specifications of the plane. Typical figures are 100 mph and 10 ft per sec per sec deceleration and the kinetic energy of the ship can readily be established.

In bus and truck service, it is somewhat more involved to establish the kinetic energy, due to the varied types of brake service. For instance, there is the continuous brake service in descending grades as compared with frequent intermittent stops in the city type of service. Variable top speeds and average speeds are involved and of par-



■ Fig. 18 – Brake test results

ticular importance is the length of time during which the energy is absorbed by the brake drum. As yet, the procedure is not satisfactorily developed by which the braking energy can be set up on a time basis so that all types of brake service can be assigned a rating constant. The continuous energy absorption in a uniform downhill braking operation is much simpler to analyze than the intermittent type of brake service.

Laboratory tests have already established a figure of 1200 ft-lb of kinetic energy per sq in. of drum swept area per sec as a rate which will minimize drum heat checking and ensure brake operating temperatures within the stable operating range for friction materials.

This method of rating brakes has been presented in some detail by J. G. Oetzel.[1] It includes the element of time of brake application and possesses possibilities as a means of translating engine horsepower or potential braking horsepower into a direct function in the formulation for establishing brake capacity. At the present time, the values have not been adequately investigated or considered outside of the laboratory, so that this method of rating brakes is not ready for general usage. It is, however, one that has many possibilities for future analysis of vehicle brake capacities.

Summary

As originally stated, the purpose of this paper was to review the various methods which are the basis for rating commercial vehicle brakes at the present time. Other methods than those discussed are used in analyzing braking on other types of equipment, such as clutches for well drilling rigs, shovels, and drag lines. All of these methods are important as they indicate the complete problem in setting up formulation. The most urgent need, however, is for an understanding of the principles of rating brakes in their application to automotive vehicles. It is believed that the best results will be obtained at the present time with the following methods of rating:

1. Where pressure-area or vehicle load-area factors are used as a basis for rating brakes, the *projected* area of the liner should be used.

2. The work-factor basis for rating brakes conforms more closely to the theory of utilizing the brakes as energy-converting units. This rating method gives a better interpretation of the various designs than other methods in use. It also emphasizes the value of additional heat dissipation in the brake design.

3. With any of the methods of rating, the adequacy of the brakes for each axle should be determined on the basis of load on the axle at the ground.

In conclusion, it is urged that serious consideration be given to the evolution of a universal method of rating brakes. Such a method for determining the capacity of brakes must consider the units from an energy-converting hypothesis with factors introduced to allow for engine horsepower and type of operation. This is an educational program requiring the concerted efforts of the automotive brake industry under the guidance of the SAE.

[1] "Postwar Brakes," by J. G. Oetzel. Presented at a meeting of the Metropolitan Section of the SAE, New York City, May 10, 1945.

DISCUSSION

Stresses Importance Of Weight Transfer

—F. M. AMBLER
Mack Mfg. Corp.

ESPECIALLY impressive are the data presented to show that the optimum arc length of brake lining is approximately 96 deg. Most spectacular is the surprisingly large reduction in air pressure required when the lining arc extends through an angle of only 90 to 95 deg: 68 psi for 116 deg of arc and only 48 psi for 92 deg of arc.

It seems very reasonable to expect that any two-shoe brake of similar design would closely duplicate these results. If this reduction in application pressure with consequent reduction in size of power chambers or cylinders does provide adequate brakes, there is not the slightest doubt that lining lengths will eventually be shortened regardless of the method used to rate brakes.

Mr. Super's paper did not mention any of the limiting factors controlling the maximum arc length on current design brakes. Referring only to two-shoe brakes, arc length is actually limited somewhat by considerations of lining to drum clearance. At each end of the lining arc a certain minimum clearance is required when the brakes are released to ensure free-running drums. If the arc length is increased the cylinder travel or cam lift required to obtain this minimum clearance also increases. Furthermore, it increases rapidly. Thus, extremely long arc linings reduce the service life between adjustments because a larger part of the available travel of the brake cylinder or chamber is required to close the clearance gap. This consideration should and does prevent the use of much above 120 deg of arc length.

Distribution of Forces

Brake rating would not be complete without a consideration of the distribution of the retarding forces between axles, and the effect on the rating of the brake. Although the requirements for lining area may be fulfilled, the performance of the brake will not be satisfactory unless the proper amount of retarding force suitably distributed is provided.

The proportioning of retarding force is subject to disagreement among automotive engineers due to difference of opinion regarding the importance of weight transfer. On two-axle vehicles, the allowance made for weight transfer during a stop varies from nothing up to 10% of the gross vehicle weight. The actual weight transfer

depends on the location of the center of gravity and on the rate of deceleration.

If the distribution of retarding force between axles is based on dynamic rather than static weight, it seems reasonable to expect that the proportioning of lining area should also be based on dynamic weight. Lining area distribution ought to be very close to brake power distribution.

It would seem, therefore, that a brake rating method could not be universal unless agreement is reached on the subject of weight transfer.

Limitations Given For Work Formula

—J. GEORGE OETZEL

Warner Electric Brake Mfg. Co.

LINING area, in itself, is certainly no criterion of the heat performance of a brake. Unit lining pressure, whether figured on the basis of projected area or radially, is useful to the brake designer but is no measure of the heat ability of a brake.

The work formula may be as useful when one has a sufficient background of field experience with it on which to base comparison, but it does not get down to the fundamental principles of heat absorption and dissipation. As a matter of fact, it represents the torque per cubic inch of swept volume. If the numerator, weight times rolling radius, is multiplied by the road coefficient – say 0.7 – it becomes the torque at wheel slide. If the denominator, brake diameter squared times width of shoe, is multiplied by 0.7854, it becomes the swept volume of the brake.

This is an index of the size of the brake relative to the work it must do, but the ratio of torque to swept area would be a more direct index of heat ability. It is the working surface, or swept area, that must absorb and dissipate the heat. Heat dissipation may be directly from the working surface as well as from the outside of the drum, so a factor representing the exposed inner surface will be needed as soon as it can be determined.

The data shown are very interesting. I am inclined to think that the increased operating pressure for liner arc lengths greater than 90 deg is a matter of geometry and relative deformation of drum and shoe. The small differential of temperature between inner and outer drum surface needs more explanation.

The wear curve seems to bear a relation to surface temperature. As the liner arc length is reduced the unit rate of heat generation is increased. Surface temperature is a function of unit rate of heat generation, and evidently a critical temperature is reached at about 86 deg, at which point the lining surface begins to disintegrate.

Our tests have indicated that surface temperature is proportional to weight squared, times speed cubed, times rate of deceleration and, obviously, this is per unit of lining area. The rate of heat generation is not uniform throughout the length of lining arc but is related to the radial pressure, or pressure normal to the friction surface, at each increment of arc.

Coming back to the curves of temperature of inner and outer surface, I would first question if the temperature of the inner surface is actually that as it leaves the point of lining contact, or if it is the temperature of a point some distance below the surface which represents the average of the period of (a) absorption and conduction through the drum and (b) direct radiation from the working surface. At high unit rates of heat generation the temperature gradient is very high, so that a very thin layer of the surface of the drum must absorb the heat. Probably the difference between the measured inner drum temperature and the actual temperature of the surface in contact with the lining is vastly greater for the shorter than for the longer lengths of lining arc.

I mention this because of its relation to wear. The binders used in linings are mostly hydrocarbons, and hydrocarbons begin to dissociate at about 750 F. Rapid disintegration of the lining surface, rapid wear, would, therefore, begin at about 750 F. In discussing radiation from the inner surface, Mr. Super mentions an inner surface temperature of 1100 F, which is quite possible. J. V. Bassett, Raybestos-Manhattan, Inc., has indicated that a temperature of 1800 F was necessary to reproduce the metallographic structure found on many drum surfaces.

Our tests indicate that it may be possible to establish a definite relationship between wear rate and the surface temperature factor mentioned above.

Speed of Vehicle Suggested As Factor in Rating Brakes

—B. E. HOUSE

Bendix Products Division

WE agree that using gross vehicle weight per square inch of lining as a basis for determining brake performance is a very superficial criterion. This is apparent when one considers that a 16-in. diameter x 3-in. wide brake has the same lining area as an 8-in. x 6-in. brake, yet the latter produces only one-half as much torque as the former for the same input. Nevertheless, the gross vehicle weight per square inch of lining is the same in each case.

We do show the above factor on our brake analysis sheet, and separately for each axle; however, we do not use it as a sole criterion to predict brake performance.

Simply adding more lining does not always increase the capacity of the brake. Sometimes, however, lining is added, for instance, to promote interchangeability or conservation of pedal travel by reducing length of unsupported drum surface between the shoes. Since this results in a compromise between cost and performance, the brake engineer must appraise its value.

The choice between cost and performance might well be one way for power-operated heavy truck and bus brakes, and the opposite way for physically operated light truck brakes.

Drums for power-operated heavy truck and bus brakes are usually of comparatively heavy section so that drum distortion is minimized. What travel losses do occur from this source represent losses in travel in the power operating mechanism, which usually is not serious. Since brake diameters are large and linings are rather wide with the power-operated truck and bus brakes, the volume of lining not worn off may be appreciable, and this waste, from a monetary standpoint, might be considerable. Therefore, the choice of short linings for power braking is indicated.

Some light trucks use drums of rather light construction, so that drum distortion might be considerable. In a physical brake hookup, because of limitations imposed by the hookup ratios required, the pedal travel available is just barely sufficient to prevent complete loss of pedal due to drum expansion and other losses in high-speed braking. Additional losses due to severe drum distortion caused by a comparatively large proportion of the drum being unsupported by the linings might result in insufficient pedal reserve. Such severe drum distortions also make the brake pedal feel "rubbery." In this case the volume of brake lining waste introduced by lengthening the lining to improve the performance is small and is justified. Therefore, the long linings are most suitable for physical operation.

In rating brakes by the *projected* area method, Mr. Super indicates that the gross vehicle weight per square inch of lining must be increased by the ratio of the arcuate length to the projected length. This ratio varies from 1 to 1 for very short linings, to 1.57 to 1 with linings 180 deg long. This, of course, assumes that the linings are positioned symmetrically about the geometrical axis of pressure, that is, a line perpendicular to a diameter drawn through the anchor pin. Since most of our brakes have linings approximately 120 deg in length, it is immaterial to us whether we say 40 lb gvw per sq in. of *arcuate* area, or 50 lb gvw per sq in. of *projected* area; the final results are the same.

Neither the "gross vehicle weight per square inch of lining" method nor the "pounds brake applying force per square inch of lining" takes into consideration the speed of the vehicle.

Since the kinetic energy of the vehicle increases as the square of the speed, and since vehicle speeds today are quite high, this is an important factor and should be taken into account when rating brakes. Therefore, we feel that a rating based on energy absorption per unit area of lining per unit of time is a necessary consideration.

In determining this value, we calculate the kinetic energy of the load on each axle, based on the maximum speed, and calculate the stopping time for the maximum deceleration required. Knowing the lining area per brake, the maximum kinetic energy in foot-pounds which each square inch of lining must absorb in one second is calculated. In our calculations we aim for 1200 ft-lb per sq in. per sec; however, satisfactory results have been obtained with values in excess of 1600.

The Changing Bus Fleet Maintenance Picture

by FLOYD PATRAS
Southwestern Greyhound Lines, Inc.

THE AUTHOR: FLOYD PATRAS (M '41), who has been a maintenance expert for over 20 years, is now manager of maintenance for Southwestern Greyhound Lines, Inc., a position he has held since 1933. He was formerly affiliated with the Mesaba Motor Co. and Northland Greyhound Lines.

WE are told that nothing is permanent except change, and nothing can more truly be said of fleet maintenance during the past four or five years.

Prior to the turmoil which followed the Pearl Harbor attack, most fleet operators had worked out what seemed at the time to be fairly satisfactory operating and maintenance practices. Some operators felt that because of the small size of the fleet or the small number of vehicles of a certain type it was advisable to replace vehicles at frequent intervals. Some others found that farming out most or all of their maintenance work probably was the best course to follow, while others maintained or attempted to maintain all of their vehicles in their own shops.

War emergency shortages struck all alike, but found the operator who had his own maintenance establishment to be in the least difficult position. We don't know how the first two above-mentioned operators managed to keep operating but have a vivid recollection of a number of the problems faced by the latter, who had a maintenance establishment.

When mushrooming war industries commenced to operate, it suddenly seemed that everyone in America had decided to travel; so buses were overloaded frequently to as much as 180% of the load for which they were designed. These loads had to be carried by parts that were worn far beyond the previously accepted wear limit, or by reconditioned parts, or by new parts, many of which were made of substandard material.

As travel increased, the demand went up for more miles per coach per month, and at the same time the replacement parts situation deteriorated and the skilled labor situation became more and more critical. Not only did we lose men to the armed forces and to defense plants, but it became necessary to divert more and more mechanics from regular maintenance work to the job of reconditioning and remanufacturing of used parts.

Fuel, as we all know, became less and less satisfactory and blowby and dilution did all sorts of things to our engine oils.

Slower operating speeds were productive of several kinds of difficulties, among which were the abuses to which were subjected such coaches as were underpowered for even normal loads in hilly country. Holding speeds down on descending grades caused drivers to take everything they could get out of their vehicles when climbing hills. Slower road speeds, of course, resulted in over-cooled engines, fouled plugs, sludging, and other ills, such as poorly heated coaches, which were equipped with hot water heating systems.

Arrangements with the operating department brought about more careful dispatching with shorter layover time at outlying points, resulting in more miles per month per vehicle.

The parts shortages problem has not been solved by reconditioning and remanufacturing, and won't be solved by anything except a bountiful supply of new parts.

Many parts reconditioning methods were tried with varying degrees of success. The following is a brief summary of most of them: Acetylene and arc welding processes

THE extensive reconditioning of parts that was practiced during the war will slowly be cut down as new parts again become plentiful. Then, Mr. Patras explains, the deciding factor will be cost. The rule as he practices it is: Recondition used parts wherever satisfactory service can be obtained from the reconditioned part and the cost of reconditioning is equal to or below the cost of a new part plus freight and handling.

In addition the author suggested that improvements were particularly required in certain specific items, such as steering gears, oil seals, oil pumps, and oil pressure regulating valves.

He explained further that if designers were to utilize to the greatest extent possible units and parts already in volume production, the inventory problems of the modern motor bus operators would be greatly simplified, and the cost of repair parts could be just that much less.

The shortage of skilled mechanics is also discussed by Mr. Patras, who suggests that the fleet operator himself establish a comprehensive training program or assist in some form of cooperative effort along this direction.

[This paper was presented at a meeting of the Milwaukee Section of the SAE, Milwaukee, Wis., Jan. 4, 1946.]

were used in salvaging such items as wheels, hubs, broken axle shafts, cylinder blocks, cylinder heads, crankcases, transmission cases, crankshafts, the building up of worn valve faces with hard surface materials, and the building up of clutch driving plates, brake camshafts, engine camshafts, distorted brake shoes, and so on, with appropriate materials.

■ Cold Welding

Cold welding was applied to heads, blocks, crankcases, and a few other parts. A few parts so repaired delivered satisfactory miles, but in most cases the repair only got us out of a temporary difficulty. This process, like many others, is satisfactory for some applications, but is not a cure-all.

Some of the low-temperature melting point alloy welding materials which have come onto the market in the last few years have worked out to considerable advantage, particularly where high-temperature welding materials serve especially well in the repair of exterior cracks of water jackets, crankcases, transmission cases, and so on; also, for building up of worn surfaces, such as brake camshafts and spring shackles. Incidentally, some of these materials provide surprisingly hard wear surfaces.

Hard chrome plating was used to build up worn piston pins, clutch driving plates, oil pump gear teeth, and a few other items. This process is quite satisfactory and made it possible to keep several vehicles on the road that otherwise would have been tied up. The cost of processing parts by this method makes it, we believe, not satisfactory for economical fleet maintenance.

Metal spraying was used very extensively and contributed very largely to our being able to keep as many coaches rolling as we did. This process was tried in all sorts of applications, even though we knew that lasting repairs might not be effected. The mere fact that we could gain even a few more thousands of miles made the effort worth while.

The spraying of hard metals on such items as worn rocker arm shafts, oil pump shafts, and water pump shafts makes such repaired parts superior to new ones.

Spraying of zinc on piston skirts made it possible to reuse worn pistons in oversize reground blocks. We were not, of course, able to redimension the ring grooves. Nevertheless such a salvaged piston was better than no piston at all.

By building up valve stems with the spray gun, we were able to reuse worn and irreplaceable valve guides simply by reaming them to a reasonable oversize and then grinding the builtup valve stems to fit.

About 8% of the metalized crankshaft pins and journals failed in service. We believe detonation due to war-grade fuels was mostly responsible.

Metalizing of rear-axle tubes and several similar items was quickly discontinued, however, when we found that the preparation for metalizing resulted in sufficient reduction of the thickness of the section of the part as to cause breakage at an early date. Metal spraying in the inside of the bores of warped cylinder blocks is a satisfactory method of bringing the bores back to original dimension and contour so that standard sleeves will fit. Spraying the outside surface of the sleeve also makes a satisfactory repair where it is not considered essential that standard outside diameter dimensions of sleeves be maintained.

Shot blasting of crankshaft fillets definitely has been found to pay off in added crankshaft miles when the shafts are shot blasted at the time of each regrinding.

It has been the practice to regrind transmission gear bores and the corresponding surfaces of the shaft and reassemble with oversize needle rollers when surfaces had become pitted by needle bearings. When oversize needle bearings were no longer obtainable, we found the use of composition bushings to be more satisfactory than the needle bearings had ever been.

The foregoing discussion might, perhaps, lead one to believe that during the war emergency we have learned so many salvage tricks that we can carry salvage and parts reconditioning operations in the postwar period to a far greater extent than was prewar practice. We actually believe, however, that exactly the reverse is true.

In our own case cost per mile of salvaging and reconditioning of parts has proved to be completely out of line with prewar use of new parts. It has always been our practice to recondition used parts wherever satisfactory service could be obtained from the reconditioned parts and the cost of reconditioning was equal to or below the cost of a new part plus freight and handling.

During the past few years our rate per hour paid to mechanics has approximately doubled, but the output per man has remained about the same per hour or per day.

We are in the process of rechecking all of our parts salvage costs and finding that it is no longer economical to recondition a large number of different parts formerly reconditioned; but that it is now much less expensive to replace with new parts as they become available.

The prices which we must pay for new parts are continually increasing; so it is necessary to recheck reconditioning costs against the cost of new parts at frequent intervals.

We are sure, however, that the vehicle designer can help us to keep our cost down and at the same time assist the manufacturer of the vehicle building up the volume of parts sales. With this in mind, we would like to make the following suggestion:

The fleet operator must obtain the maximum number of hours of service and miles of service from each vehicle every month. This can best be accomplished, of course, by means of replacement units which can be installed at selective mileage intervals or as the necessity arises. It should be possible to remove a worn unit and replace it with an overhauled and tested unit during the normal service period which is set up for each vehicle. It should never be necessary for a vehicle in a busy fleet to fail to make a scheduled departure or a payload trip because of the necessity of removing and replacing a transmission or an engine or a differential carrier or a steering gear.

Incidentally, steering gears are almost always the hardest units to get at. We surely wish the steering gear would last as long as the designer apparently thought it would when he buried it among a lot of brackets and pedals and structural members, many of which must be removed before the steering gear can be reached at all.

It is still necessary, just as it was 10 or more years ago, to tie the vehicle up at the end of every few hundred miles for a grease job. Too often it is not possible to lubricate it as often as the designer apparently thought the job should be done. Grease lubrication should be eliminated in favor of fluid lubricants wherever possible. This is particularly true in the case of wheel hubs. The hubs must be

removed entirely too frequently so that the bearings can be repacked. Furthermore, it is almost impossible to obtain a correct bearing adjustment when the bearings are packed with thick, so-called wheel bearing grease.

Much more attention surely should be given to oil and grease seals. Oil seal life is governed not by wear of the sealing lip but by the amount of abrasives thrown in between the outer surface of the seal and the metal shaft or the companion flange hub or whatever the seal may bear against. After the seal is damaged in this way the abrasives work into the lubricant and soon ruin the bearing that the seal is intended to protect; therefore, the seal should be designed to seal both ways to prevent the loss of oil and to prevent the admission of foreign matter from the outside. Two-way oil seals are particularly necessary in wheel hubs. Water enters the hub and ruins the lubricant whenever the vehicle is forced to traverse flooded highways for considerable distances, as frequently happens in the Southwest. Therefore, it is particularly necessary that a seal be developed and used that will not only keep the oil in the hub but will keep the water and other foreign material out.

Much more attention should be given to the control of engine oil temperatures. Provision should be made for the quick warm-up of the oil and then the temperature should be controlled between rather narrow limits. All of the so-called oil control devices or oil coolers with which we are familiar clog very easily and are extremely hard, if not impossible, to clean thoroughly.

Oil pump capacity should be greater in most of the engines used in heavy-duty commercial vehicles. Furthermore, the bottom pan should be equipped with one or more small, easily removed plates so that the sludge could be thoroughly removed from the oil pan at each oil change period. On all too many engines it is necessary to remove the lower portion of the crankcase completely in order to remove sludge, or, as is usually the case, the operator merely removes the drain plug and hopes that enough of the sludge may come out.

■ Oil Pressure Regulating Valve

We would like to suggest that the oiling system on heavy-duty truck and bus engines be so designed that the oil pressure regulating valve be located at the farthest point in the oil galley from the pump. When the regulating valve is located in the pump body or near the pump, we have no means of knowing that any pressure is reaching the farthest away bearings and bushings. The rapid wear of some of these points indicates that the pressure dropped off due to line losses, thereby starving certain parts which require considerable lubrication.

Most modern motor buses are highly specialized vehicles. The engine, for instance, usually is of the type which does not lend itself to many other uses. The same is quite often true of the transmission, axle, and various other units. Since the annual output of such units and their component parts is very small, the production cost is high. This is also true in the case of repair parts.

The manufacturer cannot usually anticipate the usage of repair parts and he doesn't want to tie his money up in inventories of parts to be used during the life of the vehicle, therefore he stocks only a few parts of each type. Later, when another production run must be made to take care of his customers' needs for repair parts, he must, of course, add his tool setup cost to the cost of the parts, but this time the production run is on a repair part volume only. The setup cost must be spread over a much smaller volume and the cost per unit, therefore, naturally goes up.

■ Utilization of Parts in Volume Production

We would, therefore, like to suggest that whenever possible the designer utilize to the greatest extent possible units and parts already in volume production. Perhaps it is not too much to expect that some day soon someone will sell his engines at such a low price that it will be more economical to scrap a worn-out engine and replace it with a new one than to repair the old one.

This plan might perhaps call for the use of two medium-size engines instead of one big one. Such a design, we believe, would have certain advantages. Wartime overloads have called our attention to the fact that some auxiliary power is sometimes needed badly. We can think of many reasons why twin units might be particularly beneficial in the elimination of road failures.

We would also like to express the opinion that the fleet operator should help himself by either establishing and maintaining a comprehensive training program for mechanics or by assisting in some form of cooperative effort in that direction.

Good mechanics were not plentiful before the war and they are even scarcer now.

The large number of overage automobiles that are now operating and the fact that most must be operated for many more months or years compels the service station operator to compete with the fleet maintenance manager for the few skilled men that are available. Furthermore, the average garage mechanic frequently is not the best fleet maintenance mechanic.

For some reason, mechanics during recent years have lost or discarded the pride in their craftsmanship which compelled them to turn out only top quality work.

Fortunately the armed services are now returning to civilian life large numbers of men who have had excellent technical and fundamental training as mechanics. Most of these men are eager and willing and can be made into excellent mechanics. However, they still need a great deal of training and coaching. We believe that fleet operators can profit most by employing aggressive young men. Fresh young chaps can only be obtained as they are released from the armed services. We should employ them and train them to the point that they can best utilize their initiative and resourcefulness in the fleet maintenance field.

In conclusion, we would like to say that bus fleet maintenance procedures have been and will continue to be in a continuous state of change. Rarely do two operators adopt the same maintenance procedure, and it is probably best that they do not. Each different operation and each different part of the country present problems which are different from those of an operator operating in a different location, having different types of equipment, or perhaps under longer or shorter haul conditions. Each operator, if he is to keep his cost in line, must revise and improve his maintenance procedures with regard to postwar costs and conditions.

Postwar designed vehicles, of course, will again compel us to revaluate and revise our maintenance procedure at probably more frequent intervals than was the case prior to the war.

ENGINEERING DEVELOPMENT of the JET EN

IT seems desirable that a paper at this time should be broad and simple in treatment: It should establish a groundwork and dispel current fallacies for those new to the subject; it should indicate promising lines of further development; and it should, perhaps, suggest a merit rating by which we can measure the characteristics of different combustion chambers against each other and tell when, where, and how much progress is being made.

I have not given equations or formulas, but I have tried to indicate the observation, reasoning, and analysis that should precede their development.

■ Requirements and Problems

As an airplane powerplant, the internal-combustion turbine is subject to a number of special requirements:

1. It should start easily, positively, and consistently, without detriment to engine life; and it should restart in the air without complex or difficult manipulation by the pilot.

2. The engine should fire, without blow-out or die-out, at all speeds, altitudes, and throttle positions that the pilot can use in flying. All air/fuel regulation should be automatic, and it should not be necessary for the pilot to nurse or feel the throttles.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 9, 1946.]

3. To ensure safety from fire risk, ignition should be positively accomplished under any rate of fuel flow, and the burner construction should be such that liquid fuel will not accumulate nor seep through the engine.

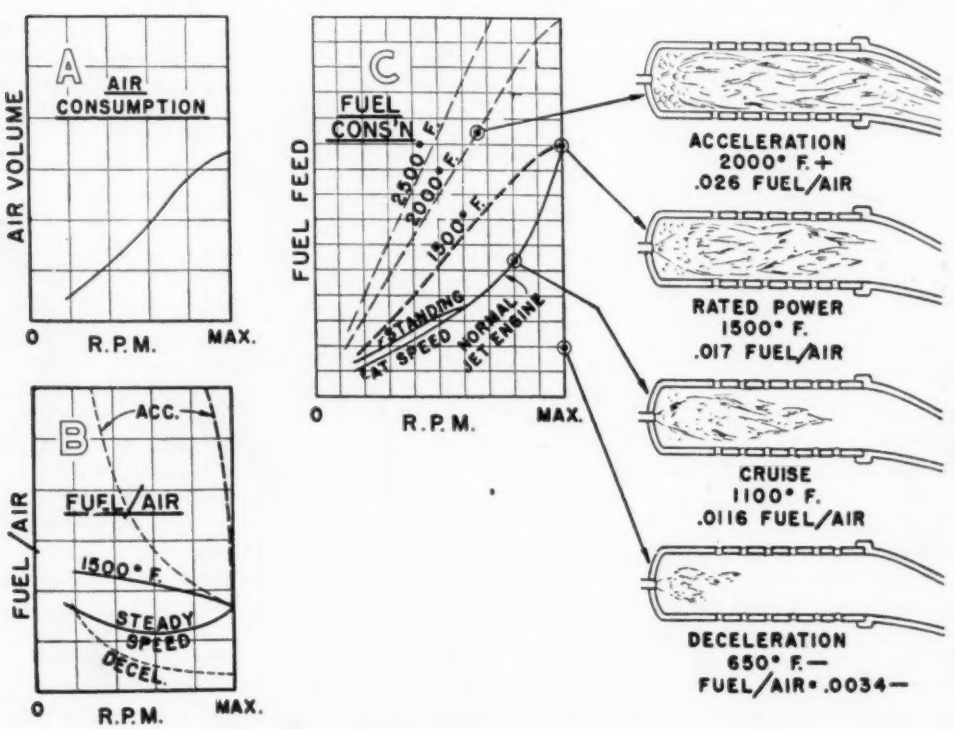
4. Carbon accumulation should be the minimum in the burner and particularly on the spray nozzles.

5. Since for many duties relatively volatile fuels may have to be used, precautions must be taken against the several forms of vapor lock.

The foregoing are, of course, in addition to the universal powerplant requirements of efficiency, complete combustion, proper temperature distribution, and the usual aircraft demands of minimum volume and weight.

We already have good evidence that all these can be achieved.

Since the gas turbine stands or falls according to the efficiency of its compressor-turbine system, this latter has received most attention, charting, and analysis. The elements involved in combustion, less often considered, are shown in Fig. 1. Depending in detail upon the compressor and turbine characteristics, the *air volume consumption* of an engine varies, very roughly, as the engine speed, as shown on Chart *A*. The temperature rise due to combustion should, and probably would if you measure it in enough places, vary with the fuel/air ratio. Taking into account the temperature of compression, we get, in Chart *C*, the *diagonal fuel feed lines of constant burner*



■ Fig. 1 – Characteristic relations involved in jet engine combustion-chamber study

JET ENGINE and GAS TURBINE BURNER

by FRANK C. MOCK
Bendix Products Division
Bendix Aviation Corp.

DESIGNING a jet engine, Mr. Mock warns at the beginning of his paper, is anything but a simple matter.

In addition to the usual powerplant requirements of efficiency, complete combustion, proper temperature distribution, and minimum volume and weight, he points out that the internal-combustion turbine is subject to certain special requirements for aircraft operation, which he outlines as follows:

1. It should start easily, positively, and consistently, without detriment to engine life; it should restart in the air without complex or difficult manipulation by the pilot.

2. The engine should fire without blow-out or die-out at all speeds, altitudes, and throttle positions that the pilot can use in flying. All air/fuel regulation should be automatic; it should not be necessary for the pilot to nurse the throttles.

3. To avoid fire risk, ignition should be positively accomplished under any rate of fuel flow, and burner construction should be such that liquid fuel will not accumulate or seep through the engine.

4. Carbon accumulation should be minimum in the burner, particularly on the spray nozzles.

5. Since for many duties relatively volatile fuels may have to be used, precautions must be taken against vapor lock.

Although the author does not claim that all of these requirements have been achieved as yet, he feels that in time they certainly will be. As evidence, he reveals that clean, compact combustion, wide range of inflammability, and positive starting under such adverse conditions as, for instance, 1200 to 1 total air/fuel ratio has already been accomplished in his own laboratory.

In addition to suggesting a detailed comparative rating of merit for different burners to enable experimental workers to appraise their models against other American and foreign types, the author sets forth in simple, easily visualized forms, the essential factors of jet engine combustion, and gives some idea of the broad possibilities of improvement that exist in the use of graduated air velocity, controlled turbulence, and impact heating spray.

THE AUTHOR: FRANK C. MOCK (M '11), whose automotive career began in 1903, has spent much of that time in research. Joining Bendix Aviation Corp. in 1930 in charge of engine research, he has since been promoted to manager of aircraft carburetor engineering and sales. He started research engineering in 1912 with Stromberg Carburetor Co., and during World War I went to the U. S. Bureau of Standards for altitude and carburetor research.

temperature, with 1500 F assumed as the safe upper limit.

The *fuel feed actually required* at different engine speeds *with a straight jet engine* is shown, for steady speed, in Chart *C;* these values at the low end run considerably higher when standing still than in high-speed flight. *The fuel required by a turbine-propeller engine* may lie somewhere between the "jet engine" line and the "1500 F" line. The temporary fuel feed may on acceleration lie considerably above these values, and on deceleration considerably below them.

Chart *B* shows these same conditions in terms of fuel/total air ratio plotted against rpm; this chart will be shown later, on a larger scale and with numbered ordinates.

I have, perhaps, oversimplified in these diagrams the effect of air-speed ram. There are really three effects:

(a) An increase of pressure and temperature and a net gain of *density* by the air entering the compressor and the burner. This is most marked at high engine speed.

(b) An increase in the pressure potential causing air to go through the engine, and thus increasing the *volume* of the air charge. This effect is most marked at low speeds.

(c) A wind-milling effect on the turbine and/or axial compressor shaft, which lowers the minimum self-sustaining firing speed, and raises the idling speed for a given fuel flow.

It is a characteristic of tubular burners with air holes along the side that the length of flame body varies with the fuel/air ratio, to a considerable extent independent of the actual amount of fuel being fed. That is, the flame volume increases and the fuel spray continues burning until enough air has been encountered to complete combustion. Since the burners are designed with minimum volume for safe maximum steady power and something like 1500 F upper limit, a richer mixture will give not only hot burners but also flame reaching through the turbine wheel. Conversely, the flame volume diminishes as the load and fuel/air ratios are decreased.

Now considering burner temperature as a limiting factor, together with maximum rpm, the engine should operate anywhere between the "1500 F" line (or some other safe limit) and the quantity of fuel needed under deceleration and idling. But we don't always achieve this at present. We have trouble starting; the minimum rate of fuel delivery required to establish flame gives much too high a temperature once the fire is started, and starting is a difficult and dangerous ordeal; there are various dead bands in the area of desired engine operation; the throttle must be nursed open carefully to avoid blow-out; and sometimes the engine may die out if the throttle be closed too quickly. Added to the foregoing, carbon deposit, rank smell, and smoke haze in the exhaust sometimes create the suspicion that our combustion is not as good as it should be.

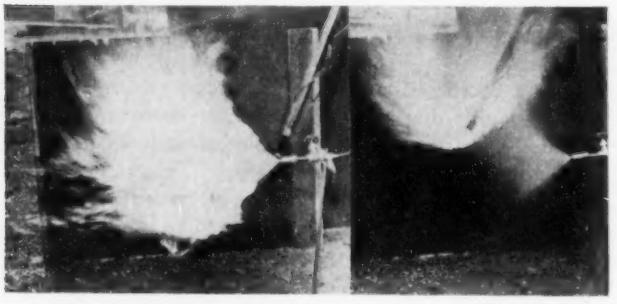
Most of us, having had casual experience with combustion all our lives, all the way from leaf bonfires to Bunsen burners and blow torches, consider ourselves experts, and we tend to think this jet burner art should be easy. To induce a proper attitude of respect, I want you to study Fig. 2. Here is shown a well-atomized jet of kerosene spray discharging into an 8- or 10-mph wind. It can be ignited, as shown on the right, from the downwind end, but the flame cannot propagate up to the nozzle; whereas, it will burn down to the nozzle tip if ignited at the upwind side. Yet in the jet engine burners we are called on to maintain stable combustion at over 80 mph air speed in the burner, 120 fps.

It is interesting to note that the fuel velocity leaving the nozzle tip is of the order of 100 fps, more or less, although within one or two feet it decelerates to the velocity of the surrounding air; and yet the flame will propagate upstream against this fuel velocity. The explanation probably is that successive droplets are thrown off the spray body and decelerate to rest in a very short space, all along the length of the spray cone. With gasoline, the flame will propagate up a solid stream of fuel, but it will not do so with kerosene.

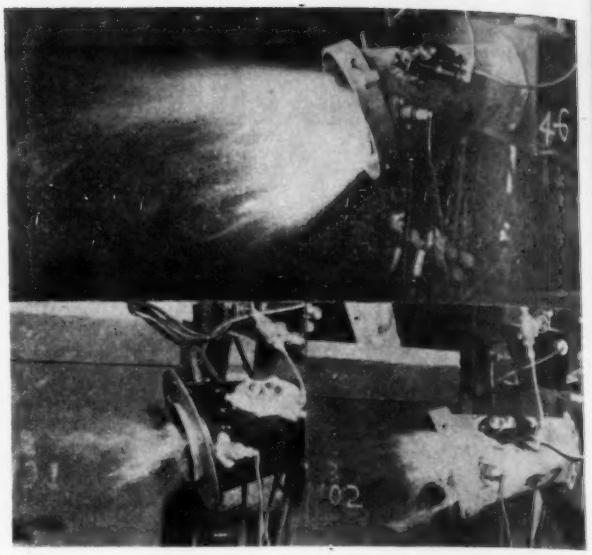
Still speaking generally, the experimenter in the first or kindergarten stage of the art will probably find the following points:

1. Maintaining a flame is more difficult in higher air velocities; and blowing more cold air at a flame often doesn't improve it.

2. At a given air velocity, flame once started can be maintained more easily with higher fuel feels, and as the body of flame is greater.



■ Fig. 2 – Well-atomized jet of kerosene spray discharging into 8-10-mph wind; left: flame is easily ignited on upwind side; right: if ignited on downwind end, flame cannot propagate back to nozzle



■ Fig. 3 – Characteristic view of flame in truncated burner; upper view: 240 lb fuel per hr normal airflow – wide spray angle striking burner near edge – stable flame; lower left: fray-out of flame just before total die-out – 33 lb fuel per hr – normal starting airflow – about 80 deg spray cone angle; lower right: fray-out just before total die-out – 20-24 lb fuel per hr – normal starting airflow – duplex spray nozzle with wide-angle spray striking dome near edge

3. In simple forms of burners, yellow, greasy flames seem more stable than white or blue ones.

4. Again in simple forms of burners, attempts to obtain more complete combustion by using lean mixtures with white or blue flames frequently result either in blow-out or in partial fray-out, with tongues of blue flame dying out before combustion is complete; accompanied by the pungent and eye-irritating aldehydes. Fig. 3 illustrates these fray-outs on the two lower pictures, along with more stable combustion at higher air flow and fuel flow in the upper picture.

5. While there is much prior burner art in oil-fired boilers, domestic heaters, and the like, it does not seem particularly applicable to jet engine service. This may be because one of the main factors of efficiency in the boilers and heaters was the radiation rate to a heat exchange wall; which radiation was so much better with red flames as to offset minor attendant inefficiencies in combustion. With the jet engine, in contrast, we want to retain as much heat as possible in the burner blast.

■ Flame Propagation

Now many of these phenomena may be characterized in terms of "normal flame propagation rate," which in this art may be defined as the maximum rate of nonturbulent or streamlined mixture flow in a straight channel under which a flame front will maintain a stable distance from a stationary point of fuel emission. Keeping as far as possible from thermochemistry, these main considerations affecting flame propagation are shown in Fig. 4. Assuming successive lamina of equal weights of mixtures of air and fuel spray, *A*, *B*, *C*, and *D* already burned, and *E*, *F*, and *G* unburned, it will be seen that in igniting *E* from *D*, the Btu liberated in the temperature rise of *D* from ignition to combustion, that is, $T_c - T_i$ is available to communicate heat to the unignited element *E*; while the Btu needed for ignition of any one laminum

equal those needed to bring its air and spray drops from their respective initial temperatures, T_{oa} and T_{oi}, up to the ignition temperature T_i; plus the latent heat of vaporization of the liquid fuel. The following considerations result:

Flame Propagation Rate Will Be Greater:

1. As the Btu of combustion per pound is greater (that is, as T_c-T_i is greater).
2. As T_i-T_o is less (that is, as T_o is higher).
3. As specific heat is less.

Flame Propagation Rate Will Be Slower:

4. As weight of inert diluents is greater (excess air or fuel will act as diluent).
5. As specific heat of inert diluent is greater.
6. As heat of vaporization of liquid fuel is greater.

The foregoing is merely a study of the heat balance necessary for flame propagation, and is seldom realized in fact. Actually, there is usually a scale effect, a big flame propagates faster than a little one. We know that the increased radiation from the big flame helps. Also, the actual propagation is borne on the expanding face of the already burned gas, and may be superimposed on whatever turbulence exists locally, whether of large or small scale (and in addition, if a chain reaction is set up, it may be imposed upon a front moving at the velocity of sound; which has been advanced as an explanation of the detonation blow-out).

In any case, one important objective of the engineer's effort to improve burner action is to increase the rate of actual flame propagation at the lower fuel deliveries. We know that the normal rate, as from *D* to *E*, is too low; we can't even fire mixtures of leaner than about 20/1 air-fuel ratio in our reciprocating engines, with their high compression and hot combustion chambers, but we do get much higher flame propagation rates than we get in many jet burners.

Many of the steps we take to improve combustion amount, in effect, to using the heat from *A-B-C* bodies of ignited gas to supplement that from *D*, in raising immediately adjacent *E* to the ignition point.

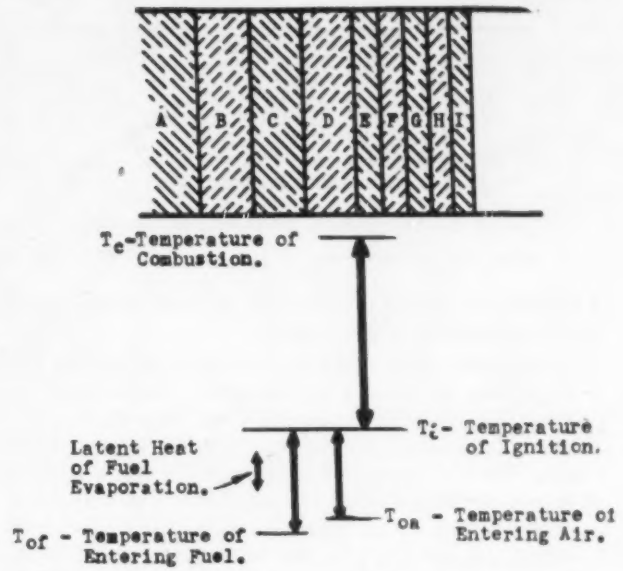
(a) We do this when we have a large body of incandescent flame, so that the radiation from its center supplements that at the outer envelope.

(b) This also occurs when the fuel is introduced first into a region of low air velocity, partly burned there, and then brought into contact with more air.

(c) It also occurs when turbulence recirculates flame back into the spray. A converse effect is generated when excess air is circulated into existing flame; if this air is cool enough and in sufficient volume, it will stop combustion and put the flame out.

(d) Finally, it is the essence of the benefit derived when the spray is discharged against a hot wall, as will be discussed later.

(e) An equivalent effect is obtained in the Junkers Jumo gas generator, as shown in Fig. 5. In this unit, a small proportion of the air enters at the front with a spiral swirl which causes it to seek the outside. Part of the fuel supply is sufficient to burn this air, the rest of the fuel being vaporized and raised to a high temperature. The combined fuel vapor and combustion products pass through flutes at the rear to the main air passage, where combustion is completed and the main temperature rise accomplished.



■ Fig. 4 – Flame propagation rate will be greater as Btu of combustion per pound is greater, that is, as T_c-T_i is greater; as T_i-T_o is less, that is, as T_o is higher; as specific heat is less. Flame propagation rate will be slower as weight of inert diluents is greater (excess air or fuel will act as diluent); as specific heat of inert diluent is greater; as heat of vaporization of liquid fuel is greater

It is interesting to note that combustion of a weight of air equal to that of the fuel charge will generate heat enough to bring a similar weight of air and the remainder of the fuel up to an igniting temperature.

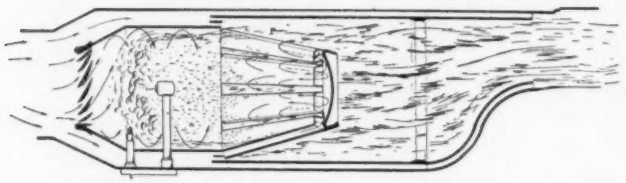
■ Controlled Application of Heat

Compression Temperature – As we saw in Fig. 4, one of the most effective factors determining the propagation rate is the initial air temperature, and this in turn follows the compression rise, and consequently, the engine speed. It is believed that one reason why critical conditions of combustion affect the intermediate- rather than the high-speed range with certain burners is this difference in compression temperature.

Variations in external atmospheric temperature will be similarly reflected in the compression temperature, and in the propagation rate.

Preheating the Fuel – This occurs at first thought as an attractive means of both raising the charge temperature and eliminating perhaps the greatest time lag in the ignition sequence, that of vaporizing fuel drops by heat from the surrounding gases. When you come to compute the actual Btu values that can be handled, even under the high limitation of coking the fuel, this method is weak. With our present system of spray nozzle regulation, with low fuel system pressures at low engine speeds, boiling limitations forbid consideration of this step.

Preheating from the Existing Flame Body – This can be done in various ways: injecting the fuel spray through a flame body that is quiescent because of limited airflow into this space; or by recirculating flame or burned gases into the region of the fuel spray, along with new air. These methods are effective, and I believe are embodied to some extent in the more successful burners in use today.



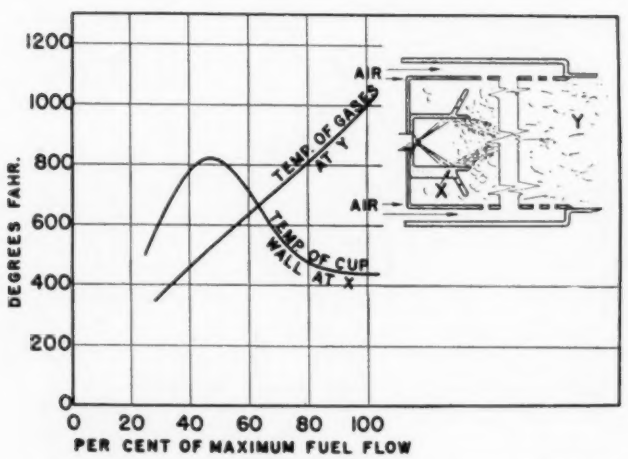
■ Fig. 5 – Junkers Jumo 004 combustion chamber

I believe that further progress will be made as we achieve better applications of this function.

Heating the Fuel Spray by Impingement against Hot Walls – This, of course, is automobile intake manifold practice; but instead of reasoning from that, we stumbled on this independently during some random burner experiments. We were surprised to find that it had a number of intriguing advantages. The construction is very favorable for starting, in the form that will be described later. It warms up quickly, and can be built to reduce materially the amount of liquid fuel lying around during the warmup period. It seems inherently to work toward a blue or intense white stable flame, with minimum fray-out and aldehydes. Fine atomization of the fuel seems unnecessary; burning is very good with a dribble from the jet. However, steps must be definitely taken to achieve diffusion, as by itself there is nothing in this vaporizing action to promote diffusion of the fuel fog through the air charge.

Possibly contrary to expectation, the difficulty is not to keep the heating surface cool, but to keep it warm enough, because the heat conductivity between a liquid and a wet wall is so much higher than that between a gas and a wall. Fig. 6 shows the general trend of temperatures with a construction like that shown. In general, the cup temperature seemed to decrease with increasing fuel flow.

It is well known, of course, that at wall temperatures above the boiling points of the fuel, some of the spray will fly off as tiny hot drops, rather than vaporizing and recondensing as it leaves the wall; nevertheless, this condition seems very favorable for flame propagation. At wall temperatures below the boiling point of the fuel, the quantity of fog will diminish and liquid drops will tear off the end of the cup. However, we have been quite surprised to see how well kerosene burns when the spray jet discharges against a relatively cool cup. One of the burners in Fig. 3 was operating in this condition.



■ Fig. 6 – Range of temperatures when fuel is sprayed against hot cup – constant airflow, variable fuel flow

As a matter of fact, this condition of liquid fuel vaporizing on a hot wall has been present, to some extent, in practically every burner form that we have observed. It is surprising how much of the time there is liquid fuel in the breech of present burners. Every now and then one of these wetted areas is adjacent to a hot wall area, and fuel steam can be seen to form as the liquid film washes along. Whether this be correct or not, we are beginning to associate this presence of fuel steam with stable blue flame.

■ Diffusion and Turbulence

To achieve the desired high liberation of energy within a small volume of combustion, it is necessary to employ a highly developed technique of air mixing, or what we may loosely call "turbulence." We can perhaps identify three steps in the sequence:

(a) Introducing enough air, rather slowly, to the initial combustion nucleus to obtain a high initial rise in temperature, but only burning a fraction of the fuel. In this operation we would particularly avoid high turbulence.

(b) Introducing the additional air necessary to approach complete combustion of the remainder of the fuel charge. This air should be introduced gradually enough so that it does not locally cool and stop flame propagation at any point.

(c) Introducing additional air to bring the flame temperature down to safe limits, say 1500 F. Obviously this step must follow closely upon (b), since we cannot have an appreciable area of chamber wall subject to the high temperatures of 15 to 1 air/fuel combustion. In this step we should also break up any concentrated tongues of flame, since these not only give localized areas of high temperature, but may also carry unburned fuel drops and bits of carbon. In this third step a high degree of turbulence may be employed, since igniting temperatures are already established.

It is thus clear that air turbulence control is one of the most vital and difficult problems in burner development. Referring first to a few elemental forms:

Air Currents Induced by Spray – These are really of appreciable force, as we should deduce from the fact brought out with Fig. 2 that fuel leaving the jet is decelerated by the air drag from 100 fps to practically nothing in less than 2 ft. These air currents, as shown in *A* of Fig. 7, are clearly marked, and are one of the chief difficulties in maintaining stable flame at low fuel deliveries. Any step such as *C*, Fig. 7, that shelters the spray cone from these currents will help stable idling, and will improve starting ignition to a marked degree. For instance, in a full-scale burner such as *A*, we could maintain flame without die-out down to a fuel flow of 7.5 lb per hr at 1775 lb airflow per hr, but when the airflow was increased 12%, we had to double the amount of fuel to keep it firing; and at 6000 lb airflow per hr, our minimum fuel feed of stable flame was 25 lb per hr. In the same main size burner we could maintain combustion with the construction of *C* through the whole airflow range at 7.5 lb fuel per hr; and I think we could have gone lower

except that this was the low atomization limit of the spray nozzle.

These may seem insignificantly low fuel-flow values, but they have an important bearing upon safe restarting in flight, as will be explained later.

Air Currents through Holes – These tend to persist for a considerable distance, and, when the air is cool, at least tend to stop combustion in their path (see *B*, Fig. 7). It is sometimes possible to see these paths in violet or dark outline through the flame. Once a high rate of combustion has been initiated, however, multiple holes give a very satisfactory air dilution and local turbulence.

The air penetration distance will, of course, be less when the openings are of such shape and size as to give high envelope surface for the area of the current, so that the viscous drag is greater.

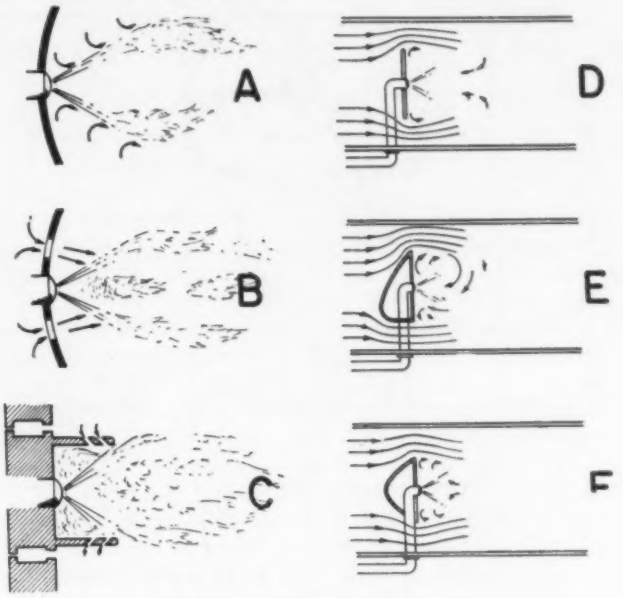
Spiral Whirls – When around the axis of the burner tube, these tend to localize the flame in the center, with a denser, cold air envelope on the outside. Mixing action does not always seem present; and for some reason, spray drops have always seemed unwilling to diffuse evenly over the outer wall; instead, the fuel collects in a stream at some erratically located point.

Eddies around Baffles – This construction is of particular interest because it is an elementary example of graduated turbulence. As indicated in *D*, Fig. 7, the region in the center of the baffle plate is relatively quiescent, particularly at the lower air speeds; though there is quite a strong local current at the lee edges. If the spray jet and ignition are located at the center, the flame stability may be somewhat better than that of Fig. 7, *A*, if the latter is in a tubular burner. At large fuel feed rates, the volume of gas flowing out from the center will tend to balance the inflowing or reverse axial air currents, thus reducing the tendency to blow the flame away; but this favorable condition will not be present in starting or idling at high air speed.

The reverse air whirls are not particularly stable behind a plate, and may change irregularly. *E* of Fig. 7 shows how to maintain a one-way rotation, and *F* of Fig. 7, how to make more stable the double-whirl type. These points may be recognized as taken from carburetor practice.

Graduated Air Velocity – A more completely controlled graduation of air velocity around the jet is shown in two forms in Fig. 8.

The two forms are quite closely equivalent. In both, a central cup is used to give a central stable flame nucleus fed only by relatively gentle axial reverse currents. Surrounding this are several annular sheaths of air at successively increasing velocities. The inner one may be delivered at a pressure differential of ⅛ in. of water (corresponding to 15 fps) when there is 2¼ in. of Hg pressure difference or 215 fps across the burner tube. The next ring of orifices may deliver at 1 in. of water pressure difference, giving 40 fps; while additional stages of increasing velocity discharge may be added as desired.

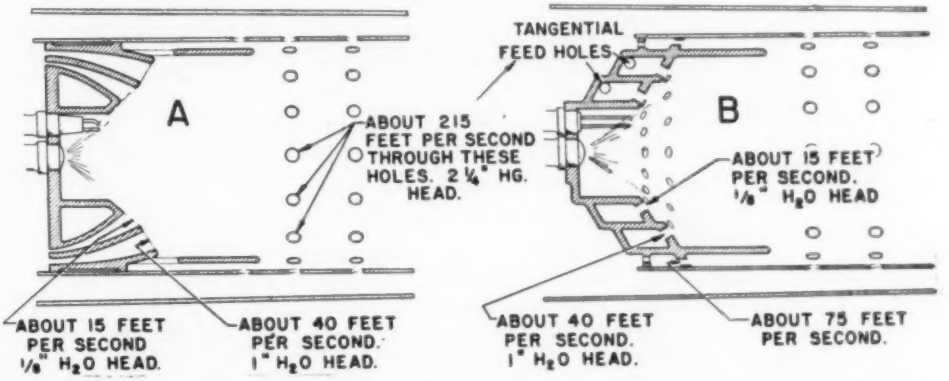


■ Fig. 7 – Various burner elements

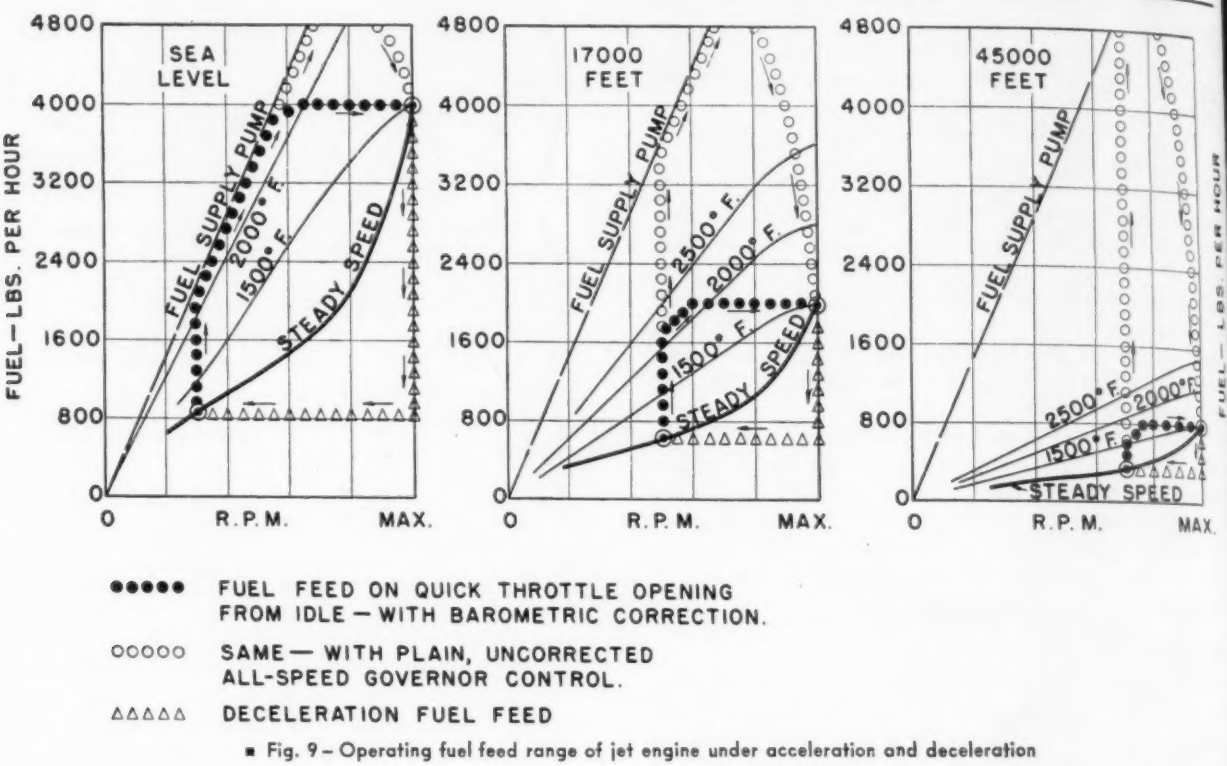
The velocity graduation in *A* is obtained by expanding conical passages (note that a larger round on the entrance will make a venturi tube out of the passage, and defeat the purpose of the construction).

In *B* the velocity graduation is obtained using a small restriction in the outer wall of each annular chamber to reduce the flow and the pressure potential across the inner orifices. With both these forms, additional side holes must be arranged downstream in the burner tube to admit more air and break up the central tongue of flame.

There are, doubtless, other ways of obtaining graduated velocity potentials around a jet. Our experience indicates that this expedient is very valuable in giving a stable flame


A
ABOUT 215 FEET PER SECOND THROUGH THESE HOLES. 2¼" HG. HEAD.
ABOUT 15 FEET PER SECOND ⅛" H_2O HEAD.
ABOUT 40 FEET PER SECOND. 1" H_2O HEAD.
TANGENTIAL FEED HOLES
B
ABOUT 15 FEET PER SECOND. ⅛" H_2O HEAD
ABOUT 40 FEET PER SECOND. 1" H_2O HEAD.
ABOUT 75 FEET PER SECOND.

■ Fig. 8 – Examples of graduated air velocity burners



●●●●● FUEL FEED ON QUICK THROTTLE OPENING FROM IDLE — WITH BAROMETRIC CORRECTION.

○○○○○ SAME — WITH PLAIN, UNCORRECTED ALL-SPEED GOVERNOR CONTROL.

△△△△△ DECELERATION FUEL FEED

■ Fig. 9 – Operating fuel feed range of jet engine under acceleration and deceleration

over wide ranges of air velocity and fuel feed rate; that the burning conditions are much more favorable, so that complete combustion can be obtained in smaller volume of burner; and cleaner, less oily flame can be used without blow-out.

Additional requirements needed, in the writer's opinion, on the forms of Fig. 8 are: first, means of heating the cup so that the spray may be allowed to impinge upon its walls; and second, development of proper turbulent mixing beyond the region shown, and addition of cooling air. Obviously, if complete combustion is obtained in a region of only 15/1 air/fuel ratio, the local temperature there will be very high, so that we cannot tolerate a very long expanse of chamber at this air/fuel ratio. As a matter of fact, the construction of Fig. 8 works best when air less than 15 times the fuel weight is admitted through the chambers shown, so that air from the side holes in the tube is required to complete the combustion.

Please note that these constructions are not the whole story: This paper was intended only to analyze and review components, not to synthesize them into a recommended complete structure. That is the problem of the development engineer, working within the framework of the engine design before him, whether it have multiple tubular burners, annular burners, or a combination of the two like the BMW-003.

■ "Detonation" of Blow-Out

This usually occurs at high air speeds and rich fuel/air ratios. It ranges all the way from an irregular exhaust sound accompanied by irregular flame length and oily smell, to a complete blow-out of one or all sections of the combustion system.

I doubt if the phenomenon is fully understood by anyone. It is known that a flame fed by a pipe carrying illuminating gas may "sing" and respond to a musical tone. Practically all large gas burners make a noise, which usually has the characteristic of irregular impulses superimposed upon one fairly steady basic frequency. I understand that quite steady flame is obtained when a combustible gas and air are thoroughly mixed in proper proportion, then allowed to issue from flame damping orifices at velocities within the propagation range, the actual flame then being beyond these orifices.

In the jet engine there are a number of special circumstances that may promote the initiation of detonation blow-out.

1. The compressor system may surge.

2. Rumble in the intake ducts of the airplane or burble at their openings may also set up periodic oscillations of pressure in the burner gas passages.

3. If the normal steady velocity in the burner system is critically near the flame propagation limit, the oscillations of (1) or (2) may take the velocity momentarily "over the edge" in the midpoint of their cycle, with resulting blow-out.

4. A chain reaction may develop in the process of combustion.

Altogether, here is a good place for a combination of basic and applied research. However, just as physicians find cures for diseases long before their nature is fully understood, we can perhaps find methods of successful attack on this problem.

1. It is known that blow-out is more apt to occur as the air/fuel ratio approaches maximum power, that is, 15 to 1: that in the jet airplane it occurs most under quick throttle opening at altitudes; and that with present con-

trols, rich mixtures are developed under acceleration at altitudes, as shown in Fig. 9. These show, on a common fuel feed ordinate, the relation of rpm to, respectively, (a) the normal fuel requirement for steady running; (b) the fuel feed corresponding to 1500 F, 2000 F, and 2500 F burner temperatures; (c) the capacity of the fuel supply pump, which with some controls is the limit of fuel feed on acceleration; and (d) the limits of fuel feed on acceleration of the usual barometric correction *set at its best;* and when there is no limit, as with an uncorrected, all-speed governor control. The need for fuel/air and temperature limit on the acceleration fuel feed is obvious, not only from the blow-out consideration, but also in view of the dangerously high temperatures, from which only blow-out now saves us. The remedy in this direction is a different control system.

2. In our tests, blow-outs seem more frequent as there is more liquid fuel in the burner breech; most people will be surprised to hear that there is even a liquid deposit on burner walls during high-power operation, but really we find it quite often. We have found that, provided the burner flame is not too oily, a beam of sunlight reflected up into it will illuminate the interior in such a way that you can see right up inside and observe both liquid fuel and unburned fog that are present. This fuel is warm but not vaporized during the steady pressure condition; but it could conceivably be vaporized during the low pressure period of a single cycle; which would give a momentary, rich mixture. Or it could conceivably be drawn along into the flame, also with a momentary enrichment. Anyway, I believe this point will bear looking into.

3. Since the blow-out is an inertia phenomenon in a pipe, it should have a defined period, and therefore, should be subject to damping by a resonator chamber. It may not be impossible to fit such a chamber on the breech of our burners; the technique might be like that of fitting the silencer chambers on our motor-car-engine air cleaners.

4. Another possible cure might be a "flame anchor" (for which term I am indebted to Mr. Nerad of the General Electric Co). We have repeatedly seen a pocket of flame, usually in a point of low air velocity, reignite the fuel fog after a blow-out. In fact, this must be what happens during the rough firing I mentioned a few paragraphs before. This is a patch, rather than a cure, of course, but if it could be brought under control, it might be worthwhile having.

Starting and Idling

The basic requirements for initial ignition are not too complex:

1. The fuel droplets must be subdivided; it is hard to ignite a liquid stream or sheet of kerosene.

2. Where the fuel is relatively nonvolatile, such as kerosene, the spark must meet the droplets; and a strong spark of large gap is desirable.

3. The fuel and air velocities must not be so high as to blow out the initial flame.

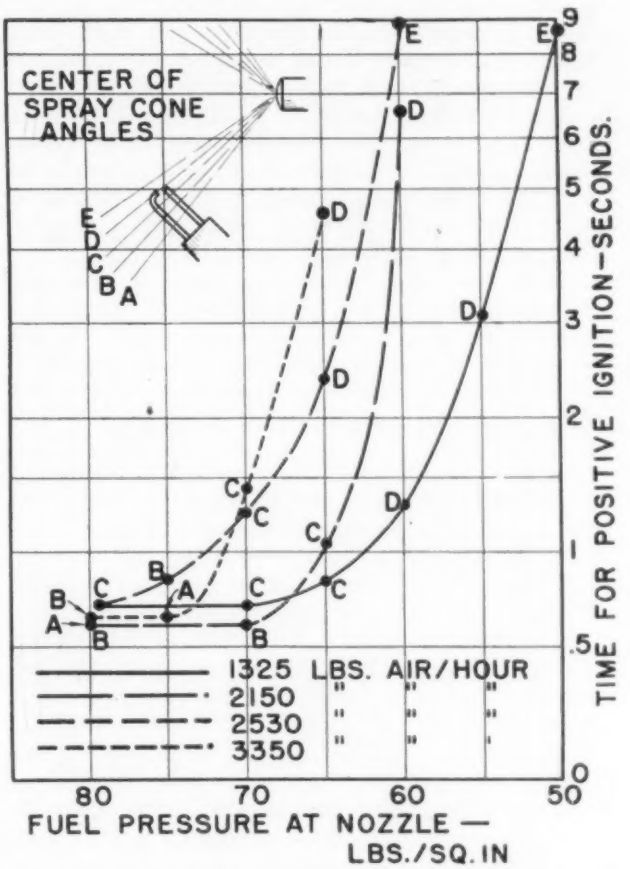
In starting a jet turbine engine, however, other considerations appear:

(a) That which concerns the burner and involves the foregoing, namely, obtaining a permanent and stable flame with no large liquid fuel accumulation in the engine the moment the fuel is turned on at the jet and initially ignited.

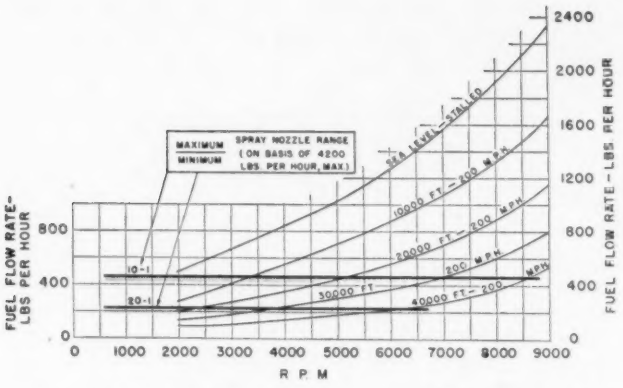
(b) That which involves the efficiency of the turbine and compressor at cranking speeds. At how low a speed can the engine develop enough power to accelerate itself, with the fuel feed limited to hold a safe burner flame temperature?

Our recent experience, working back and forth between a full-scale engine and the single burner laboratory, is that worth-while improvements as to "hot starts" may be made by improving the initial ignition capabilities of the burner. Fig. 10 shows the record of a single burner experiment in terms of airflow on the complete engine, nozzle fuel pressure, and time to obtain ignition with kerosene; also the corresponding location of the spray cone with reference to the spark gap. The use of a logarithmic ordinate has no significance; it was merely used to spread the points on the chart.

It will be noted that so long as the spray cone was just slightly outside the spark gap, ignition conditions were favorable. When the spray cone was directly in the region of the points, ignition was delayed, and when it came inside the points there was no ignition at all; there was, of course, some flickering of the spray, particularly at the low nozzle pressures and at the higher airflows, which made the reading less consistent. The airflow modified the position of the cone somewhat, as can be seen by comparison of the points in the 65-lb nozzle pressure range; but the major factor seemed to be the spray cone angle itself. The above experiment checks broadly with our experience on the complete engine.



■ Fig. 10 – Effect of fuel spray position relative to spark gaps in determining time period required for ignition



■ **Fig. 11 – Probable minimum firing limits at different altitudes as affected by rpm and spray nozzle characteristics**

Factors limiting starting at altitude:

1. Least fuel-flow rate that will cause spray to strike spark-plug gaps—below this, liner loads with fuel without igniting
2. Least fuel-flow rate that will keep flame going—partly spray nozzle characteristic, partly combustion chamber
3. Minimum self-sustaining speed of engine, say 3500 rpm; stalled, say 2500 rpm in glide at 200 mph
4. Maximum windmilling speed, with no fuel feed

Of equal importance is provision for consistent positive sparking, with provision against rapid carbon deposit and against general flooding with fuel in the region where the points issue from the insulator.

Restarting in flight is a highly complex subject; as indicated in Fig. 11, it involves such factors as:

(a) Air consumption at different speeds under windmilling glide conditions; this determines the amount of fuel that can be used without passing the safe upper limit of burner temperature.

(b) The minimum self-sustaining speed of the turbine at this upper limit of temperature, under glide conditions.

(c) The ability of the combustion chamber to maintain flame when the self-sustaining speed and combustion-chamber airflow are high.

(d) The low delivery limit of the spray nozzle *that will give adequate atomization for stable flame;* the atomizing ability of the spray nozzle is a matter only of the fuel rate through it, and it makes no concessions to altitude.

(e) The control of the spray nozzle delivery as regards striking the ignition plug spark gaps, as previously mentioned.

We have no means of telling just what the fuel requirements are for combustion at different gliding speeds at altitude, and they probably vary widely; nevertheless, their mutual relation is probably something like the guess shown in Fig. 11. Similarly, I do not know the minimum self-sustaining speed under all of these different conditions, nor the windmilling speeds that can be attained. Nevertheless, it does seem clear that with either a low-range spray nozzle or a low-range combustion chamber, restarting would be very difficult at altitude.

(a) If, in order to get adequate atomization at the spark gap, fuel is fed at a higher rate than called for by the curves of the chart, a very hot start seems inevitable.

(b) If the fuel is fed at the rate called for by the curves, but this results in a mere dribble from the nozzle, the engine can load with fuel without igniting.

(c) If the pilot windmills the engine at a high speed in order to avoid overheating from a high nozzle feed, there is danger with some combustion chambers that the high air turbulence will prevent initial ignition.

Therefore, it seems to me that we should make it our objective to obtain a combustion chamber, spray nozzle, and spark location that will ignite and keep firing at quite low limits of fuel flow and high air/fuel rates; then we would at least release the pilot from the dread of raw fuel loading in the engine and leave it up to him to dive the airplane to get the once-stalled engine up to a self-sustaining speed. All this of course involves a burner temperature limit on the fuel metering system.

■ Rating of Merit of Different Burners

At the present stage of jet engine burner development, when there is wide-spread activity under some secrecy, and when conditions do not promote general competitive test, it seems to me that there is particular need for some sort of detailed comparative rating. We need some way in which a given group can appraise their experimental models against English, German, and current American types, and estimate their progress accordingly.

Therefore, to initiate discussion of the subject, the following rating is suggested, without particular comment or defense:

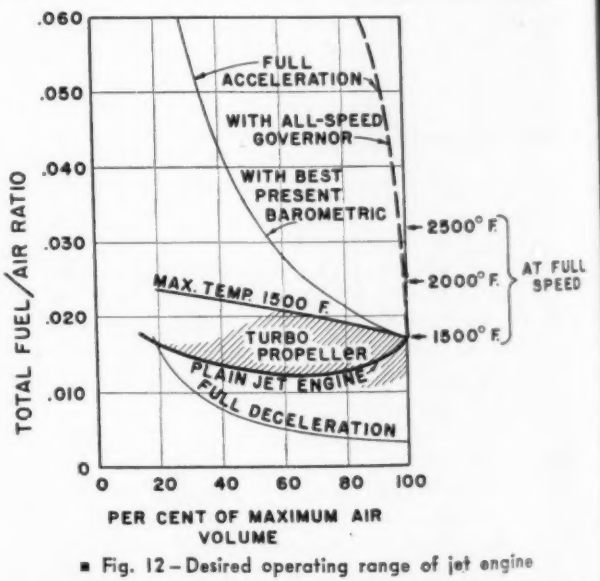
I. Range of stable combustion, perhaps plotted as combustion ability on a chart such as Fig. 12. Note the areas of essentially steady operating range, and the areas imposed by the acceleration and deceleration characteristics of the control to be used.

II. Efficiency of combustion in cruising and in the power range.

III. Excellence of fuel handling, as evidenced by carbon accumulation, rapidity of burner warmup, and by wetness of burner walls and engine during exigencies of service operation.

IV. Starting ability—to include:

(a) Minimum fuel and airflow requirements, as shown in Fig. 10. (Ability to ignite small fuel flows at high airflows will probably indicate altitude restarting excellence.)



■ **Fig. 12 – Desired operating range of jet engine**

(b) Burner temperature of steady operation, in the very low speed range; on a plot that coordinates with (a) foregoing.

V. Variation of foregoing points—I, II, III, and IV—at altitude (when it can be ascertained).

VI. Burner pressure drop at standard maximum rated airflow.

VII. Burner size or volume, per unit of air volume handled.

VIII. Sensitivity of the burner to change in fuel volatility.

Summary

In the foregoing I have tried to set forth, in simple, easily visualized forms, the more essential factors of jet engine combustion; and to give some idea of the broad possibilities of improvement that exist in the use of graduated air velocity, controlled turbulence, and impact heating of the fuel spray. Some of you have visited our laboratory and have seen what can be accomplished in the way of clean, compact combustion, wide range of inflammability, and positive starting under such adverse conditions, for instance, as 1200 to 1 total air/fuel ratio.

As previously stated, it was not my objective to announce or recommend any particular burner form. This depends too much upon specific engine design, and is too broad a subject for inclusion in this study. I do feel, however, that this subject has not received sufficient attention thus far; our engines are not aircraft engines, and our jet aircraft are not military service aircraft, unless combustion is positive and dependable under operating conditions. Therefore, the combustion chamber should be one of the first elements to be experimentally proved before a given engine arrangement is adopted.

It is hoped that this paper may assist our industry in such efforts.

DISCUSSION

Suggests Two More Characteristics For Rating of Merit Consideration

—LLOYD E. BERGGREN

Wright Aeronautical Corp.

Preheating the Fuel—On paper this method of increasing the rate of fuel vaporization may appear to be of doubtful value; however, by actual tests we have been able to improve the combustion efficiency of a burner operating on unleaded (ANF-22) gasoline 2-5% by preheating the fuel to 300 F. Undoubtedly, various burner designs will respond differently to fuel preheat, and it is conceivable that the mixing and vaporizing processes in some burner designs are so effective that the burner will not appreciate preheat at all.

Spiral Whirls—It should be pointed out that this type of flow acts as a centrifuge on the spray drops, promptly depositing them on the walls of the burner. Unless the surface on which the fuel is deposited is purposely designed to heat and thus vaporize the fuel, such practice defeats, in part at least, the purpose of the initial fuel atomization.

Rating of Merit of Different Burners—In rating any burner designed to operate in a jet engine or in a gas turbine, we would suggest that at least two more burner characteristics be considered in addition to the eight listed by Mr. Mock, namely:

(1) Burner discharge temperature pattern, expressed as a percentage variation from the mean. Inadequate mixing of primary and secondary air will reduce somewhat the turbine efficiency since it was designed to operate on a gas of uniform temperature. Furthermore, localized heating resulting from poor discharge temperature patterns may adversely affect turbine durability.

(2) Burner durability:

(a) Structural.

(b) Ability to maintain initial combustion efficiency. This point is mentioned because it is true that in some existing burners efficiency decreases with running time as a result of structural deformations, atomizer wear, and/or carbon deposition.

Results Given for Another Type of Combustion Chamber

—D. G. SAMARAS

National Research Council of Canada

GENERALLY speaking, the two most important requirements of a combustion chamber are a high combustion efficiency and a low pressure loss, although small bulk and weight are universal requirements of aircraft elements.

From tests made in the Royal Aircraft Establishment, it was found that these two requirements were interchangeable. For a given weight and bulk of combustion unit, a high efficiency is achieved at the expense of high total head pressure losses and vice versa.

It has been found that in a combustion chamber, two limits might be investigated, usually the upper or flame extinction limit and the lower or ignition limit.

The flame extinction limit is defined as the limit at which combustion can still continue, and the ignition limit as that at which combustion can still be started.

Measurements made in an experimental combustion chamber show that the ignition limit lies considerably lower than the flame extinction limit.

Both limits show a considerable drop in velocity in the neighborhood of the stoichiometric mixture. The more favorable variation of the upper limit is attributed to the higher temperatures present in the primary zone as compared to ignition during the starting process. It is also seen that by increasing the temperature the lower limit is improved.

Absolute pressure has little effect on the upper limit and temperature has a marked higher effect.

If the fuel pressure is increased, better atomization will result and the flame extinction limit will be higher.

Back pressure of the combustion chamber has a definite influence on both the upper and lower limits. The effect of back pressure results in great difficulties at high-altitude operation.

Regarding the starting problem, from thermodynamic considerations it is seen that a low total head pressure loss accompanied by high combustion efficiency is highly desirable, as are high compressor and turbine efficiencies at low rpm.

It was found that the velocity of the flame propagation with turbulent and laminar flow varied as $W_{ft}/W_{fe} \propto \sqrt{Re}$ and, for higher Reynolds numbers, was proportional to Re. This may be considered as one of the chief reasons why a big flame propagates faster.

As the velocity of flame propagation increases in turbulent flow, the total head pressure loss also increases.

In an engine combustion chamber a considerable degree of turbulence is produced by the compressor but still more is required in order to secure high combustion intensities, especially in the secondary zone.

By "blow-out" the author means the flame extinction limit. In an unpublished work by the author the criterion of flame stability was found to be that of thermal equilibrium (heat balance).

1500 F is quoted as the upper limit. This is the present-day upper limit for uncooled turbine blades; for cooled blades this can be increased as high as 2000 F, depending on the amount of cooling air.

Author's Closure To Discussions

I fully concur with the comments of Mr. Berggren.

Major Samaras has evidently misunderstood the author's meaning at several points; which is not surprising since this art is so new that it has neither adequate nor generally accepted vocabulary. For instance, the term "turbulence" may be applied either to the small-scale, irregular motion of the flame front, as referred to by Major Samaras, or to the large-scale, confused motion in the burner chamber resulting from the inward "squirt" of air through ½-in. holes both in front of and behind the flame front. The two conditions are, of course, quite different.

Nevertheless, Major Samaras' data and theories are most welcome as showing the results obtained with one type of combustion chamber.

NEW Synthetic LU

by

J. C. KRATZER
Linde Air Products Co.,

D. H. GREEN
National Carbon Co., Inc., and

D. B. WILLIAMS
Carbide and Carbon Chemicals Corp.

THE purpose of this paper is to describe in some detail the characteristics of two series of synthetic lubricants having certain unusual properties. Until recently these compounds were subject to an order of secrecy issued by the Commissioner of Patents. Both series of lubricants are currently being produced on a commercial scale by Carbide and Carbon Chemicals Corp. One series of synthetic lubricants, which is essentially insoluble in water, is designated by the symbol "LB" followed by a number representing the viscosity at 100 F in Saybolt universal seconds. The other series, the members of which are soluble in water at room temperature, is designated by the symbol "50-HB" similarly followed by a number designating the Saybolt viscosity at 100 F.

These synthetic lubricants were developed by units of Union Carbide and Carbon Corp. The original work was carried out at the Mellon Institute of Industrial Research by the Organic Synthesis Fellowship, which is sponsored by Carbide and Carbon Chemicals Corp., in the early 30's. Further development of these products has since then been a continuous effort, not only by the Organic Synthesis Fellowship, but in the laboratories of the Carbide and Carbon Chemicals Corp. at South Charleston, W. Va., and in the laboratories of the Linde Air Products Co. at Tonawanda, N. Y., where automotive research work in connection with antifreezes and other automotive products has been conducted for many years. Important contributions have been made to this development by H. R. Fife of the Organic Synthesis Fellowship at Mellon Institute, W. J. Toussaint and Dr. F. H. Roberts at South Charleston. Evaluation and development of these materials for engine use was conducted at the Tonawanda laboratory of the Linde Air Products Co. Important contributions at that laboratory were made by E. E. Sommer, L. J. Clapsadle, and Dr. G. H. Wagner.

These new synthetic lubricants are a development of American science and industry. They are not related to the synthetic lubricants developed in Germany. Those were primarily developed to substitute for natural petroleum products which were unavailable to the Germans during the war period. They exhibit properties which are superior to these German products, and they differ from them chemically.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 7, 1946.]

The new synthetic lubricants to be described here are synthesized from natural or other hydrocarbon gases as raw materials. They contain no petroleum oils. They are inherently more expensive to produce than the best petroleum lubricants now sold. This factor will probably limit the market for these products except where their special properties justify increased cost. Both the LB series (water-insoluble) and the 50-HB series (water-soluble) of synthetic lubricants are being manufactured and sold by Carbide and Carbon Chemicals Corp. under the trade-mark "Ucon" for a wide variety of industrial applications.

■ LB Series

The LB series which, because of its special properties, is best adapted for the lubrication of machinery in general, including internal-combustion engines, will be considered first. Physical data for some of the common viscosity grades of lubricants of the LB series are given in Table 1.

These compounds may be prepared in any desired viscosity by controlling the reaction employed in the manufacturing process. It is not necessary to blend higher and lower viscosity oils, as is the general practice in pre-

Table 1 – Properties of Ucon Brand Lubricants – LB Series

		LB-140	LB-300	LB-400	LB-550	LB-650
Viscosity, SUS at	210 F	45.5	62.7	74.3	91.9	107
	100 F	140	300	400	550	650
	0 F	4960	18,380	27,600	40,800	50,500
	−30 F	42,100	183,000	264,000	444,000	540,000
	−50 F	265,000	1,370,000	2,335,000	3,890,000	—
Viscosity, centistokes at	210 F	5.9	11.0	14.1	18.5	22.0
	100 F	29.8	65.0	86.6	119	141
	0 F	1080	4000	6000	8900	11,000
	−30 F	9200	40,000	62,000	97,000	118,00
	−50 F	58,000	300,000	510,000	850,000	—
V.I. (ASTM D-587-41)		147	142	142	140	140
Pour Point (ASTM D-97-39), F		−50	−40	−35	−30	−25
Density at 210 F, g per cc		0.921	0.933	0.936	0.939	0.940
Density at 100 F, g per cc		0.966	0.979	0.983	0.985	0.986
Specific Gravity at 60 F		0.983	0.997	1.001	1.003	1.004
Gravity, deg API at 60 F		12.5	10.5	9.7	9.5	9.4
Flash Point (ASTM D-92-33), F		440	470	485	510	525
Fire Point (ASTM D-92-33), F		510	570	580	580	590
Carbon Residue (ASTM D-189-41), %		Less than 0.01				
Ash (ASTM D-482-43-T), %		Less than 0.01				

Table 2 – Piston-Ring Wear Data

Lubricant	Average Piston-Ring Gap Increase, in.
Oil B-1	0.0060
Oil B-2	0.0069
Oil B-3	0.0065
Oil B-4	0.0064
Lubricant LB-300	0.0061

LUBRICANTS

DESCRIBED here are the detailed characteristics of two series of synthetic lubricants, one essentially insoluble and the other soluble in water at room temperature.

These lubricants are synthesized from natural or other hydrocarbon gases as raw materials.

Results of laboratory and field tests indicate that the lubricants can be successfully used in internal-combustion engines. In fact, the authors report that one of them has already been used extensively in aircraft engines by the Army Air Forces and the Air Transport Command.

■

THE AUTHORS: J. C. KRATZER (M '44) has been continuously associated with the Research Laboratory of Linde Air Products Co. since his graduation from Washton State College with a master's degree in mechanical engineering in 1936. He has recently been connected with the development of the new synthetic lubricant for use in internal-combustion engines. D. H. GREEN (M '38) who has been with National Carbon Co., Inc., for the past 27 years, is now manager of the organization's automotive engineering department. Active in transportation and maintenance activities of the Society, Mr. Green is a member of the SAE-Ordnance Vehicle Maintenance Committee on Limits and Tolerances for Replacement of Parts and Units, and of the SAE War Engineering Board Antifreeze Committee. D. B. WILLIAMS (M '30) joined Carbide & Carbon Chemicals Corp. upon graduation from Cornell University in 1924, and has been with the company ever since. As manager of the Special Products Division, he is concerned with the development and sale of new synthetic lubricants.

paring certain SAE grades from neutral and bright stock in the oil industry. However, blends of the synthetic products can be made if desired. The high viscosity indexes, characteristic of both series of lubricants, are inherent and not dependent on viscosity index improvers.

The low API gravities of these materials, coupled with high viscosity indexes, stamp them as being chemically different from petroleum oils. These synthetic lubricants contain no wax or any additives to depress the pour point. The pour point as determined by ASTM method D-97-39 is a stable pour and can be reproduced after repeated temperature cycles. Flash and fire points compare favorably with mineral oils, although in the lower viscosity range, below LB-140, the flash points are higher than for equivalent viscosity petroleum oils. Auto-ignition temperatures are about 100 F above petroleum oils of equal viscosity. The low carbon residue values exhibited by these synthetic lubricants are probably indicative of the low carbon formation we have observed when they are used to lubricate internal-combustion engines.

The oxidation products of the synthetic lubricants are, for the most part, low molecular weight volatile compounds, while in the normal process of oxidation of mineral oil, nonvolatile products are formed which eventually convert to oil-insoluble sludges, gum, and varnish. In the case of the synthetic lubricant, insoluble oxidation products which tend to foul an engine are not formed, and this factor is responsible for negligible increase in the viscosity of the lubricant with use.

Two of the lubricants listed in Table 1 are particularly important and have been selected for discussion because their viscosity characteristics cover the lubricating grades most generally recommended for automotive and aircraft use. These are lubricants LB-300 and LB-550. Ucon lubricant LB-300 has been selected for general automotive use since its viscosity is equivalent to most SAE 10 oils at 0 F, most SAE 20 oils at 130 F, and most SAE 30 oils at 210 F. Ucon lubricant LB-550, which is equivalent in viscosity to most SAE 50 oils at 210 F and is lighter than most SAE 30 oils at 0 F, has been widely used by the Army Air Forces and the Air Transport Command in aircraft engines. This use will be discussed later in the paper.

Table 3 – **Average Wear Data from Standard 36-Hr Chevrolet Test Using Lubricant LB-300**

Average Piston-Ring Gap Increase, in.	0.0061
Average Rod Journal Diameter Decrease, in.	0.0002
Average Main Journal Diameter Decrease, in.	0.0002
Average Wristpin Diameter Decrease, in.	0.0001
Average Copper-Lead Half-Bearing Weight Loss, g	0.137

Table 4 – **36-Hr Chevrolet Oxidation Test on LB-300 Lubricant and Oil B-3**

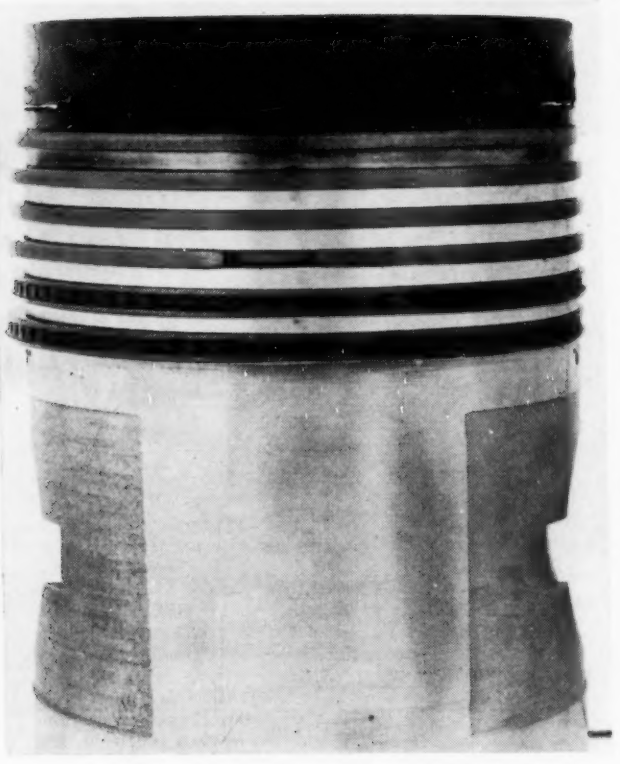
	LB-300	Oil B-3
Engine Cleanliness Rating:		
Varnish	46.7	47.1
Sludge	46.1	46.9
Total	92.8	94.0
Cu-Pb Bearing Weight Loss per One-Half Bearing Insert, g	0.137	0.210
Oil Viscosity Increase, %	8.0	25.0
Deterioration:		
Free Acid, mg per g	0.012	—
Total Acid, mg per g	0.050	—
Ash, %	0.240	—
Neutralization No.	—	0.65
Conradson Carbon	—	1.90

Table 5 – **Average Weight Loss per One-Half Copper-Lead Bearing Insert**

Lubricant	Weight Loss, g
Oil B-1	0.302
Oil B-2	1.050
Oil B-3	0.257
Oil B-4	0.127
Lubricant LB-300	0.137

■ LB-300

Lubricant LB-300 was being marketed by National Carbon Co., Inc., a unit of Union Carbide and Carbon Corp., in two limited areas in the East the winter of 1946 for use in automobiles, buses, trucks, and tractors. These selected areas were used because available production facilities were limited and distribution had to be localized. National Carbon Co. is calling the product Prestone motor oil,



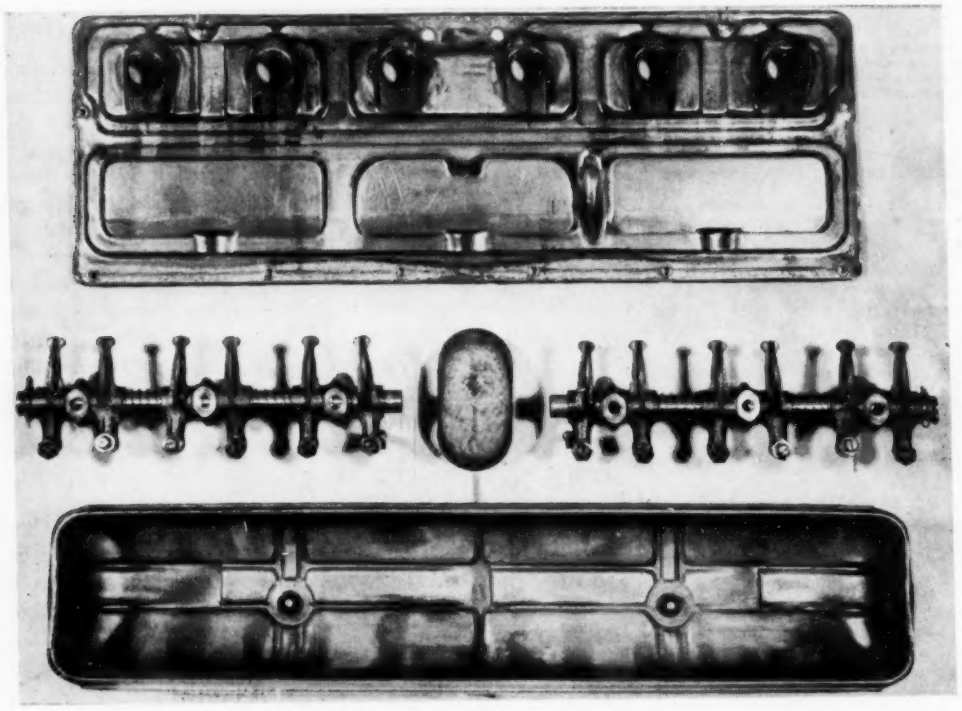
■ Fig. 1 – Caterpillar scratch test piston

sharing the trade-mark of its well-known antifreeze. National Carbon Co. is offering Prestone motor oil for sale only in these areas until increased production facilities become available.

In considering a product which is chemically, and in other respects, quite different from petroleum oils, a number of questions arise: Is it a good lubricant? Is it resistant to oxidation? How does wear of the engine compare with the best petroleum oils? Does an engine consume more oil? Does it corrode the bearings and other parts of the engine? Does it permit engine starting under low-temperature conditions? Does it form more or less sludge and varnish in an engine than petroleum oils? The Tonawanda laboratories are well equipped to provide the answers to these questions and work along these lines has been in progress for several years. In addition to a fleet of laboratory owned test cars, 40 vehicles operated by laboratory employees have been continuously used for test purposes. Also, over one hundred automobiles operated by private individuals have been on a test basis for over a year. Road experience has been further supplemented by tests in three privately owned truck fleets having no connection at all with Union Carbide and Carbon Corp. or its subsidiaries. The experiences of the Army Ordnance Department and the Army Air Forces have added to our knowledge of the performance of these products in engines.

The Tonawanda laboratory has eight engine dynamometers. Engines available for test purposes include two Caterpillar single-cylinder test engines, one General Motors 471 diesel, Chevrolet engines, automobile engines made by other manufacturers, two COT engines, and a number of Lauson single-cylinder engines. This laboratory has participated in several lubricating oil test programs with other cooperating laboratories, for example in the work authorized by Subdivision B of the Lubricants Division, SAE Standards Committee, to develop a method for evaluating heavy-duty oils. This laboratory's test results have correlated well with other laboratories in the group.

While lubricant LB-300 and other members of the LB series have been subjected to most of the standard laboratory tests used for petroleum oils, we feel that performance in engines on the dynamometer stand and in vehicles has



■ Fig. 2 – Appearance of engine parts after completion of Chevrolet 36-hr test with lubricant LB-300

the greatest significance. Results observed when using LB-300 in engines will first be reviewed.

Wear Characteristics

Lubricant LB-300 was tested in a great many engines on the dynamometer stands and operated over two million miles in various types of automobiles and trucks before being offered for sale for automotive use. Measurements have demonstrated that wear of the moving parts is comparable to wear with mineral oils. Piston-ring wear data for LB-300 obtained from a great many standard 36-hr Chevrolet tests (CRC Designation L-4-545) run over a period of years have been averaged and are presented in Table 2 together with the average ring wear data for the Coordinating Lubricants Research Committee Oils B-1, B-2, B-3, and B-4. The wear values for the petroleum oils were obtained by averaging the data reported by all of the laboratories cooperating on this oil test program.

It is standard practice at the Tonawanda laboratory to make complete wear measurements on all the engines tested. Average wear data for the various engine parts are summarized in Table 3.

Actual wear data from road test vehicles operated under both low- and high-speed conditions for periods up to 2½ years have shown wear values comparable to those obtained with the better grades of petroleum oil. This includes vehicles with previous petroleum oil history as well as new vehicles in which nothing but the synthetic oil was used in the engines.

The Caterpillar scratch test (CRC Designation L-2-545) is recognized as a test useful in judging the antiscuffing qualities of an oil. A piston from a single-cylinder Caterpillar engine which has passed a scratch test in the Tonawanda laboratory with lubricant LB-300 is shown in Fig. 1. This test has not yet been officially conducted on this lubricant by Armour Research Foundation.

Stability and Bearing Corrosion

A low oxidation stability would stamp an oil as undesirable for use in an internal-combustion engine. Typical 36-hr Chevrolet engine test results, which are a generally accepted means of evaluating oxidation resistance of a lubricant and its corrosive effect on copper-lead bearings, are given in Table 4. Data for heavy-duty oil B-3 are presented for comparison.

The engine cleanliness rating in Table 4 is about equal to that of a good, heavy-duty mineral oil meeting Specification 2-104 (b), such as committee oil B-3. Fig. 2 shows the appearance of engine parts which were photographed after completion of the 36-hr test with lubricant LB-300.

Table 5 gives the average weight loss per one-half copper-lead bearing insert for oils B-1, B-2, B-3, and B-4 in comparison with losses when using lubricant LB-300. The weight losses shown are the average of all the results reported by the cooperating laboratories in the oil rating program previously referred to.

Special Engine Test

While on repeated tests the synthetic lubricant passes the standard 36-hr Chevrolet oxidation and corrosion test in a highly satisfactory manner, it was believed that a new type of lubricant such as this should be evaluated under more severe conditions. Therefore, LB-300 was run for

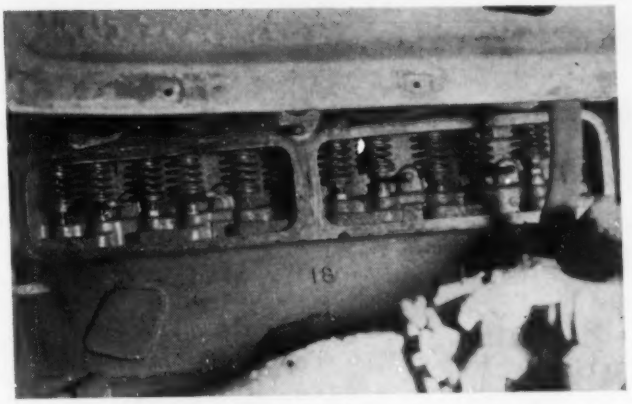


Fig. 3 – Plymouth valve chamber – 70,000 miles

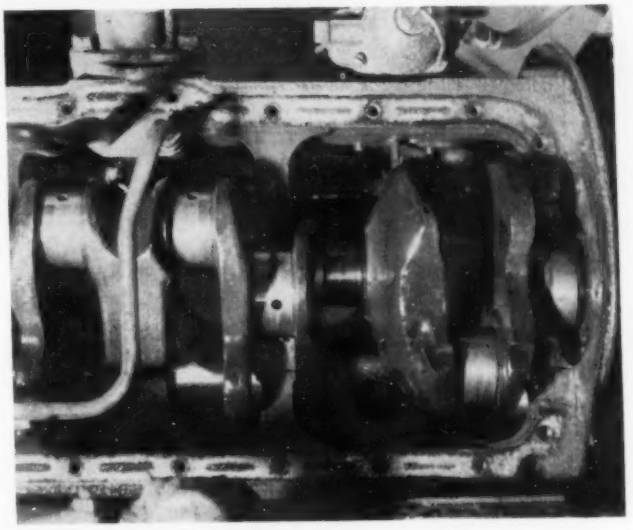


Fig. 4 – Plymouth crankshaft – 70,000 miles

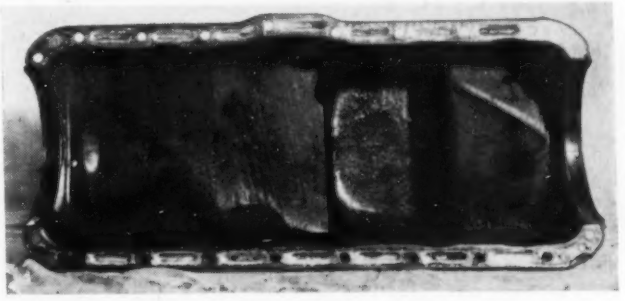


Fig. 5 – Plymouth pan – 70,000 miles

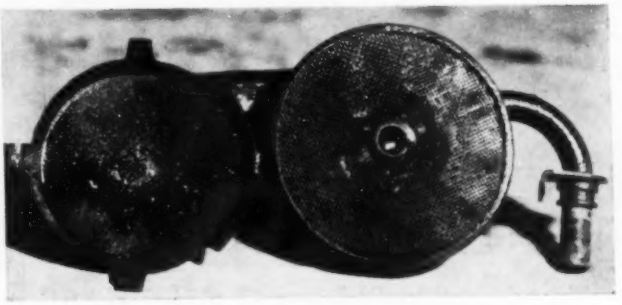


Fig. 6 – Plymouth oil screen – 70,000 miles

Table 6 – Engine Cleanliness Ratings[a]
(Varnish and Sludge)

Mileage	Engine Rating Oil B-3	Engine Rating LB-300
2,000	94.8	96.9
4,000	84.9	91.8
6,000	81.2	93.0
12,000	75.6	86.1
18,000	68.1	88.2
24,000	59.5	71.1

[a] 100% rating means a perfectly clean engine.

Table 7 – Copper-Lead Bearing Weight Loss in 24,000-Mile Test

Mileage	Average Cu-Pb Bearing Weight Loss per One-Half Bearing Insert	
	Oil B-3	LB-300
6,000	0.852	0.317
12,000	1.033[a]	0.641
18,000	1.262	1.029
24,000	1.516	1.288

[a] Renewed bearings in the oil B-3 test.

Table 8 – Lubricant Deterioration

Mileage[a]	Increase in Viscosity at 100 F, %	
	Oil B-3	LB-300
6,000	47.5	14.2
12,000	28.1	7.0
18,000	35.6	11.8
24,000	27.6	14.0

[a] Lubricant was changed at each 6000-mile period.

24,000 miles (480 hr) in a Chevrolet engine under the same oil and coolant temperatures as the standard 36-hr test (280 F oil and 200 F water), but the normal 30-hp load was increased to 60 hp and the engine speed was dropped from the standard 60 mph to 50 mph. Other engine conditions were kept the same as on the standard 36-hr test, except that the lubricant was changed after each 6000 miles of operation. In order to have some basis of comparison under this type of test condition a second Chevrolet engine, using heavy-duty committee oil B-3, was run simultaneously under exactly the same conditions. Table 6 shows the cleanliness ratings for the two lubricants under these test conditions at various mileages.

These data demonstrate that the engine remained clean for a longer operating period with the synthetic lubricant than with the petroleum oils. Since this cleanliness characteristic is inherent, this performance standard is maintained over a very long period of time with LB-300.

On this 24,000-mile test the corrosion of the copper-lead bearings was less with lubricant LB-300 than with committee oil B-3, as shown in Table 7.

The viscosity increases for the two lubricants for each 6000-mile increment of the 24,000-mile Chevrolet test previously referred to are given in Table 8.

Oil consumption data for the 24,000-mile Chevrolet test averaged 727 mpq for oil B-3, an SAE 30 oil, and 696 mpq for lubricant LB-300.

■ Field Tests in Vehicles

Extensive testing of lubricant LB-300 in vehicles under actual road conditions was conducted in a fleet of laboratory owned test cars to evaluate performance under field conditions and to obtain information with respect to service problems which might be encountered during use in engines. A number of new cars were purchased in 1942 and serviced with lubricant LB-300. Some of the engines were dismantled and the parts accurately measured prior to servicing with this synthetic lubricant, but most of the engines were started on test after a simple drain and flush procedure. This latter procedure was used so there could be no question of altering clearances while taking wear measurements. Conclusions drawn from these tests may be summarized as follows:

(a) Clean engines remained essentially sludge and varnish free under operating conditions which normally promote formation of these oxidation products.

(b) Wear of engine parts and bearings was equal to that of petroleum oils.

(c) Oil consumption compared favorably with petroleum oils.

(d) LB-300 permitted starting of the engines under low-temperature conditions.

(e) The oil does not adversely affect gaskets and fiber timing gears.

(f) Leakage is not a problem in a properly gasketed joint. However, the lubricant is a solvent for the oxidation products of petroleum oil and the cleaning action of the synthetic lubricant may tend to open up leaks plugged with oil oxidation products. While this can also result from the use of heavy-duty oils containing detergents, solvent rather than detergent action is responsible in the case of the synthetic lubricants.

(g) In cars operated exclusively on the synthetic lubricant, there was less than the normal amount of combustion-chamber deposit.

Figs. 3 to 10, inclusive, present photographs of various parts of test vehicles operated exclusively on synthetic lubricant for 50,000 to 70,000 miles. The cleanliness of the engine parts is apparent in these photographs.

■ Commercial Fleet Tests

It was realized that, while the performance of synthetic lubricant may be excellent in new vehicles, its special characteristics with respect to conditions in old vehicles should be thoroughly investigated. Therefore, several commercial fleets were selected in which the product was tested. An effort was made to provide fleets that represented conditions of (a) low mileage, high-gasoline dilution, water condensation and heavy sludge, (b) medium mileage, intermittent, and idling type of operation, and (c) heavy-duty, medium to high mileage operation under high and low ambient temperature conditions. In each of these fleets several of the engines were initially inspected and at various intervals thereafter to observe the sludge conditions. No special precautions were taken in changing these fleets over to synthetic oil except to change the filter cartridges if the vehicles were so equipped, and to change the test lubricant and filter cartridge after 500 miles of operation. Each of the fleets was run with a different oil change period and one fleet was operated without an oil change for the total mileage. In all cases the fleets were under the control of the fleet operator during the entire test. Analysis of the used lubricant samples, inspection of engines, tabulation of lubricant consumption, and analysis of the operating record constituted the test program. The significant con-

clusions drawn from these tests, which are now in their second year, are summarized below:

(a) Out of over one hundred vehicles serviced none had any loss in oil pressure indicating a clogged oil screen.

(b) Vehicles which were inspected at the start of the test and inspected periodically thereafter showed a gradual cleaning up of the engine with no formation of new sludge. After a year's operation disassembled engines were essentially free of sludge and varnish.

(c) In vehicles in which petroleum oil had been changed regularly because of lubricant deterioration as a result of severe service conditions, analysis of the synthetic lubricant indicated the change period could be at least doubled. However, in those vehicles where mineral oil was changed because of gasoline dilution or some other factor of contamination the lubricant change period could not be extended. To study resistance to deterioration a few test vehicles equipped with filters have so far been driven 30,000 miles without changing the oil. In these cases, the engines remained clean and there has been no indication of abnormal lubricant deterioration. The oil filter cartridges are being changed at recommended intervals. These tests are being continued.

(d) Improved low-temperature cranking and starting characteristics were experienced in vehicles operating in low-temperature areas.

(e) In engines where consumption of petroleum oil was high primarily as a result of leakage past the pistons into the combustion chamber, spark-plug fouling was reduced or eliminated when the synthetic oil was substituted.

(f) The average lubricant consumption compared very favorably with the consumption of petroleum oils used prior to the change. The vehicles involved in these commercial fleets were divided about equally between trucks and passenger cars. Table 9 gives the lubricant consumption data for one of these fleets.

Diesel-Engine Use

A number of 480-hr Caterpillar single-cylinder engine tests (CRC Designation L-1-545) has been run with lubricant LB-300. These tests were satisfactory except in two respects: (a) there was some impacting of carbon behind the top ring although the ring remained free in the groove, and (b) some soot from the fuel adhered so tightly to the piston skirt that light wiping with a clean rag did not completely remove it. A typical piston from a 480-hr Caterpillar run is shown in Fig. 11.

Heavy scratching of the metal which usually occurs on the piston above the ring travel area was almost completely lacking in these tests. The results indicated that some detergency would be desirable in order to pass the test, which is a requirement to meet Specification 2-104 (b), although performance of the lubricant on the 480-hr endurance run was otherwise quite satisfactory.

A small amount of detergent added to lubricant LB-300 gave improved results on the 480-hr endurance run, as shown in Fig. 12. Additional work on synthetic-base diesel lubricants is currently being conducted.

Viscosity-Temperature Relationships

The SAE viscosity classification of lubricating oils classifies in terms of viscosity only, and does not take into consideration viscosity-temperature relationship of the lubricant. According to the SAE classification system, lubricant LB-300 is an SAE 20 oil. However, because of its high viscosity index, 142, it is equal in viscosity to most SAE 10 petroleum oils at 0 F and to most SAE 30 oils at 210 F. Fig. 13 demonstrates that the product is not adequately described as an SAE 20 oil, and this one grade of synthetic lubricant covers the range of the normal SAE 10, 20, and 30 grades of petroleum oils.

The low pour point and desirable viscosity-temperature relationships ensure adequate lubrication under high- and low-temperature conditions and permit the use of lubricant LB-300 both summer and winter, making seasonal oil changes unnecessary, thereby giving better oil economy when compared with the lighter grades of mineral oil used to ensure low-temperature starting.

Film Strength

Laboratory film strength tests, for example on the Falex, Timken, and Shell 4-ball machines, have shown that lubricants of the LB and 50-HB series have film strength intermediate between straight petroleum oils and mild extreme-pressure lubricants. The 50-HB series has slightly higher film strength than the LB series. These products may be compounded with additives to produce lubricants having e-p properties. Some extreme-pressure additives used in petroleum oils are not suitable for use in the synthetics because of low solubility.

Oil Inspection Tests

Since the synthetic lubricant is chemically different from petroleum oils, many of the chemical tests applied to pe-

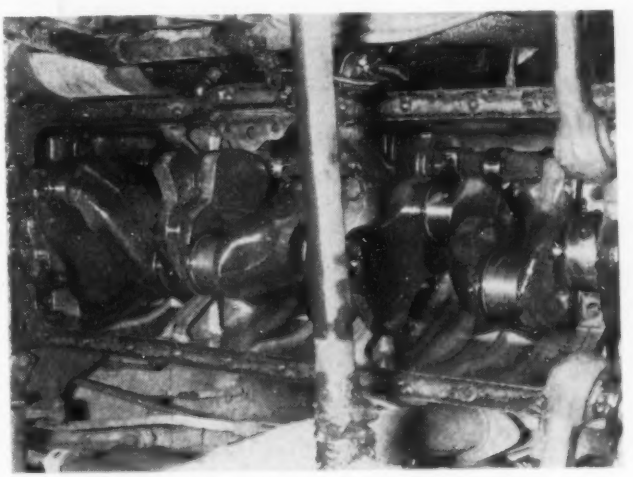
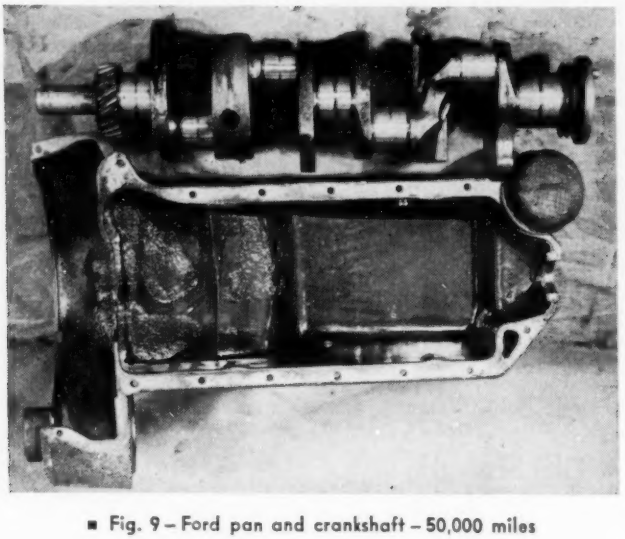


Fig. 7 – Buick crankshaft from below – 60,000 miles

Fig. 8 – Buick pan – 60,000 miles



■ Fig. 9 – Ford pan and crankshaft – 50,000 miles

■ Fig. 10 – Ford pistons and rods – 50,000 miles

troleum oils do not have the same significance. For example, the values obtained for naphtha insolubles and chloroform solubles on the used lubricant are misleading because of different solubility characteristics of the synthetic oil and its oxidation products. Neutralization number as determined by the conventional ASTM method has a different significance in the case of the synthetic lubricant. The standard neutralization number test on the synthetic lubricant is difficult to run without saponifying the esters formed, thus obtaining a value analogous to the ASTM saponification number. In view of the chemical difference between the two lubricants, we believe a more accurate method is to analyze for free as well as total acid and for this determination we use a modified ASTM method in which the results are expressed in milliequivalents per gram of oil.

The measurement of viscosity is, of course, as significant with synthetic oil as with petroleum oil, because it is a physical property.

The ash value as determined by the ASTM method appears to have the same significance as with petroleum oil since the amount of noncombustible solids in the used lubricant may have a bearing on engine wear and performance. In used synthetic lubricant, ash has been identified as largely lead from the fuel.

It is felt that viscosity, ash value, free acid, and total acid are the chemical and physical tests most useful in following the condition of the used lubricant.

Table 9 – Lubricant Consumption Data

Vehicle No.	LB-300 Vehicle Mileage	Consumption of LB-300, mpq	Previous Oil Consumption, mpq
1	8,700	432	250
2	16,000	760	180
3	4,341	877	670
4	11,000	440	396
5	8,700	458	424
6	10,250	512	334
7	3,800	3,798*	850
8	10,072	323	300
9	10,000	427	1,015
10	15,500	888	768
11	10,500	262	192
12	4,100	357	270
13	6,600	1,110	251
14	35,400	212	396
15	41,000	288	388

*Motor overhaul.

■ Solvent Action

As previously mentioned, lubricant LB-300 has a solvent action for many of the oxidation products formed from petroleum oils. The lubricant has the ability to clean most dirty engines gradually while the vehicle is being operated. This cleaning action brings up several questions concerning engine performance: (a) Will the petroleum oil oxidation products dissolved in the lubricant affect its lubricating ability? (b) Will this cleaning action loosen dirt which may subsequently be picked up by the oil pump? (c) What will happen if mineral oil is added to a crankcase containing synthetic lubricant which has a large amount of mineral oil oxidation products dissolved in it?

Taking up these questions in the same order:

(a) None of the dynamometer engine tests or field tests in vehicles so far conducted has demonstrated any detrimental action due to dissolved oxidation products of petroleum oil.

(b) There is a certain amount of dirt which circulates in the lubrication system under the most ideal conditions. It is reasonable to theorize that a larger amount, released when the gummy binders are dissolved in the synthetic, will be circulated during the period when the engine is subjected to the cleaning action of the synthetic lubricant. We feel the important point is whether the average particle size of this dirt or insoluble matter has increased. Microscopic examination has revealed that larger particles are not circulated as a result of the cleaning action of the synthetic lubricant. We have observed that any large particles settle to the bottom of the crankcase. If the oil pan is removed from a dirty engine and cleaned prior to the installation of the synthetic lubricant, much of the insoluble dirt released as a result of the cleaning action will be found in the bottom of the pan if it is again inspected after several thousand miles of operation. The question has been raised regarding the possibility of clogging with a lubricant having a cleansing action, as in the case of oils having detergent characteristics. There is no evidence from present laboratory or field test data avail-

able that the mild solvent action of lubricant LB-300 is loosening deposits that might cause or aggravate clogging. On the other hand, we have observed that in engines which had been heavily sludged previously, cleaning of the hard dense types of sludge was slow and not complete even after extensive mileages using LB-300.

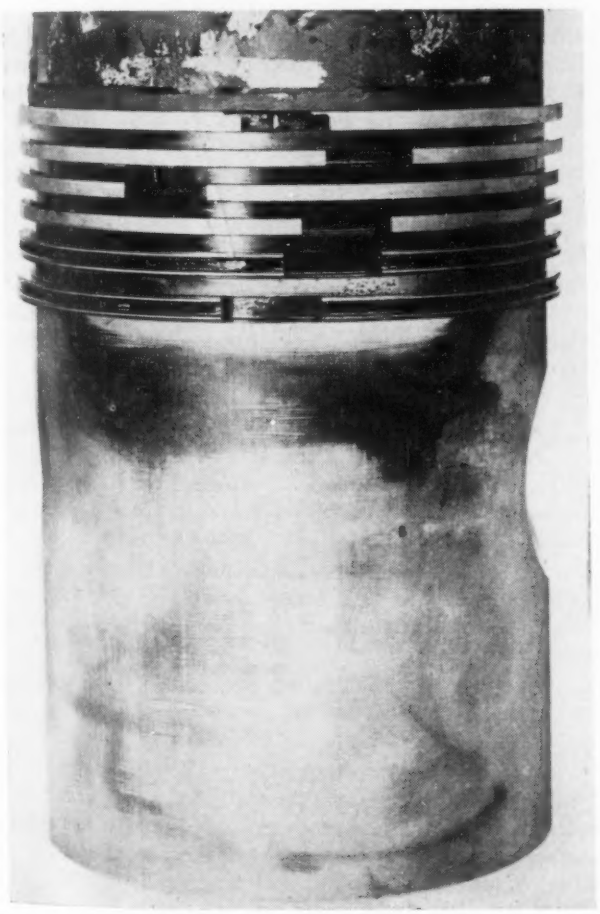
(c) If petroleum oil is added to a crankcase containing synthetic lubricant which contains a large amount of dissolved sludge and other oxidation products, a certain amount of dissolved material may be precipitated. The more oil added, the greater the amount of oxidation products so precipitated. Conversely, again adding synthetic lubricant would tend to redissolve the oxidation products. The synthetic lubricant and petroleum oil are only partly soluble in each other. We do not consider it good practice to add mineral oil to the synthetic except in an emergency, although mixtures of the two lubricants can be run in an engine free of sludge without difficulty. Tests in vehicles for 20,000 miles on a mixture of 2-104 (b) petroleum oil and synthetic lubricant have been carried out with satisfactory results.

■ Water Solubility

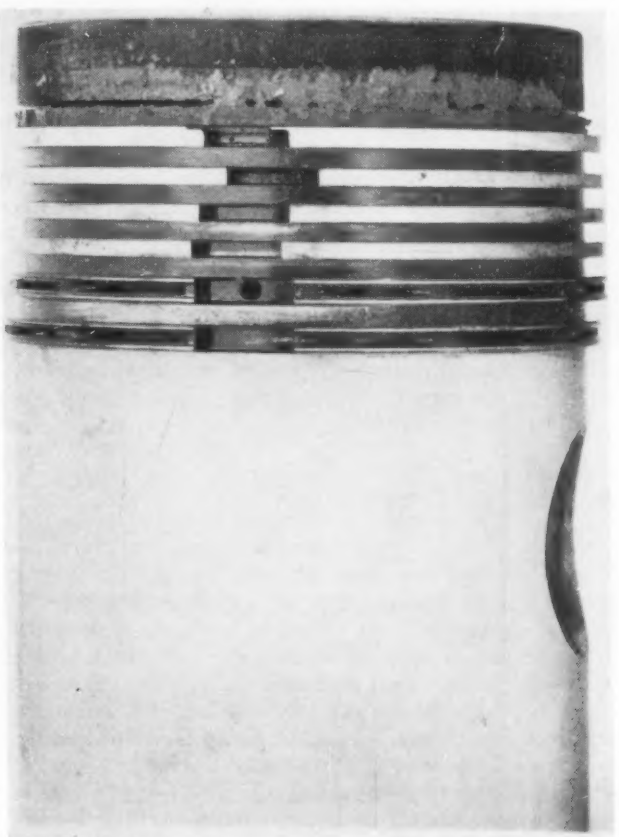
The synthetic lubricants of the LB series differ from petroleum oils in that they will dissolve and hold in solution approximately 3% of water by weight at room temperature. If water in excess of this amount exists in the crankcase, it will appear as water globules. On standing the water will form a lower layer if the temperature is above 50 F, or float on the oil if the temperature is below 50 F. This limited water solubility has certain advantages. For example, during winter operation often 1 or 2% water is present in the crankcase which remains undissolved in petroleum oil. This water, upon freezing, may present a serious lubrication problem by restricting oil flow. One or 2% water will dissolve in the synthetic and will not freeze out of solution and interfere with oil circulation even at temperatures below the pour point of the lubricant.

An operating problem which has caused considerable concern under normal winter conditions particularly in "package" delivery fleets is the control of winter sludge which contains water, oil, and oil decomposition products. While a water emulsion can be formed in an engine with the synthetic lubricant, the "mayonnaise" type of sludge is not formed, probably due to a combination of water tolerance and the absence of high molecular weight oxidation products, such as are formed from petroleum oils.

While there are some advantages due to the water tolerance previously described, it is also true that the lubricant film does not provide the same protection against humid corrosion of iron and its alloys as does used petroleum oil which normally contains protective polar compounds formed by oxidation. Iron and steel surfaces exposed to high-temperature and high-humidity conditions are protected for nearly the same length of time by the synthetic lubricant as by straight mineral oil containing no additives. However, used mineral oil provides a better protective film



■ Fig. 11 – Caterpillar piston: test L-1-545 without detergent



■ Fig. 12 – Caterpillar piston: test L-1-545 with detergent

Table 10 -- Properties of Ucon Brand Lubricants – 50-HB Series

		50-HB-55	50-HB-190	50-HB-280	50-HB-400	50-HB-660	50-HB-2000	50-HB-3520	50-HB-5100
Viscosity, SUS at	210 F	34.3	54.8	63.5	82.9	123	326	540	744
	100 F	55.0	190	260	400	660	2000	3520	5100
	20 F	315	2115	3035	5290	9430	32,650	65,300	92,000
	0 F	712	6110	9650	16,800	30,800	112,200	212,000	276,000
	−20 F	2100	27,500	39,400	66,000	122,400	550,000	780,000	1,010,000
	−40 F	7880	151,000	229,000	370,000	733,000	2,750,000	3,660,000	—
Viscosity, centistokes at	210 F	2.4	8.7	11.2	16.3	25.7	70.0	116.0	160
	100 F	8.9	41.0	56.2	86.6	143	433	760	1104
	20 F	68.6	460	660	1150	2050	7100	14,200	20,000
	0 F	155	1330	2100	3650	6700	24,400	46,200	60,000
	−20 F	458	6000	8600	14,400	26,700	120,000	170,000	220,000
	−40 F	1720	33,000	50,000	80,000	160,000	600,000	800,000	—
V.I. (ASTM D-567-41)		97	158	153	149	144	133	130	125
Pour Point (ASTM D-97-39), F		−80	−70	−65	−55	−45	−35	−35	−35
Density at	210 F, g per cc	0.920	0.970	0.977	0.985	0.991	0.997	0.997	0.991
	100 F, g per cc	0.972	1.016	1.023	1.030	1.037	1.042	1.044	1.053
	60 F, g per cc	0.993	1.034	1.040	1.047	1.053	1.059	1.058	1.059
Minimum Flash Point (ASTM D-92-33), F		255	435	435	450	450	450	470	470
Minimum Fire Point (ASTM D-92-33), F		280	480	500	500	500	500	500	500

against humid corrosion than new synthetic lubricant. The parts of a disassembled engine which has been operated on synthetic lubricant will rust quicker when exposed to high-temperature and high-humidity conditions than the parts of a similar engine operated on mineral oil for the same length of time. Over two million miles of operation in vehicles of various types under all weather conditions has demonstrated no abnormal rusting or wear of the cylinder walls and other parts of the engine. When disassembled engines operated on mineral oil are to be exposed to high-humidity conditions for extended periods of time, they should be treated with a rust preventive oil. This applies also in the case of the synthetic lubricant. Work is in progress to develop suitable additives to incorporate in the product to provide improved rust preventive characteristics.

■ Use of Filters

The use of filters when using the synthetic lubricant is desirable as the ash value can be kept at a very low point. Most of the commonly used oil filters have been tested and function satisfactorily with the synthetic lubricant. As previously explained, the viscosity increase of LB-300 in the crankcase is very low. This may have something to do with improved performance of filters in removing solids which contribute to high ash values, as increased viscosity of the oil is not a factor in preventing efficient removal of solids by the filter cartridge.

■ Lubricant Changes

Oil change recommendations for the synthetic lubricant cannot be given any more accurately than in the case of petroleum oils, as the proper oil change period depends upon a number of factors. There is, however, one essential difference which must be considered. With the synthetic lubricant the oxidation products are, for the most part, volatile, which means the lubricant remains essentially unchanged with use, but with mineral oils nonvolatile products are formed which eventually become converted to oil insoluble gums, sludges, and varnish. Beyond a certain period which depends on service conditions, these decomposition products build up in petroleum oils, necessitating a change of lubricant in order to maintain the proper viscosity. Considering only this basic factor, no change of the synthetic oil would be necessary. However, from a practical standpoint the presence of road dirt, metal particles, and so on, make oil changes necessary, and there is no reason to believe that the synthetic lubricant can be used longer when these contaminants are present than when using petroleum oils.

■ Aircraft-Engine Use

Synthetic lubricant LB-550 has been used extensively in aircraft engines by the Army Air Forces and the Air Transport Command. To date, over 150,000 engine-hours have been accumulated on this lubricant. Some of the engines operated over 900 hr before overhaul. While the bulk of the oil was consumed in Canada and Alaska, a number of aircraft were flown on the synthetic lubricant in this country during the summer of 1944. The lubricant was directly substituted for petroleum oil in these engines. As a result of this experience, the following conclusions were drawn:

(a) It was always possible in the equipment tested to start engines at ambient air temperatures of −15 F without gasoline dilution, and starts have been observed at −30 F without dilution.

(b) Freedom from sludge and varnish normally formed by oxidation of petroleum oil in the engines was clearly demonstrated. This was noticeable particularly in the piston-ring grooves which were always free from sludge deposits.

(c) The oil cooler tubes remained relatively free of sludge accumulations. Furthermore, under extreme low-temperature conditions congealing of oil in the oil coolers was not experienced.

(d) It was possible to feather successfully the propellers hydraulically operated by the engine oil at temperatures down to −60 F.

(e) The sludge formed was essentially lead compounds from the fuel. These lead compounds being uncontaminated with oil oxidation products proved to be denser than the deposits in engines operated on mineral oil, and it is felt that such hard accumulations at certain points in the engine would be undesirable.

(f) Past experience of the ATC has indicated some diffi-

culty with respect to exhaust valve sticking which resulted in engine roughness after 400-500 hr of operation of engines. This same difficulty has been encountered with the conventional type of oils but to a lesser degree based upon the test data. Valve sticking did not occur when valve guide clearances were adequate.

(g) Some increased leakage as compared with petroleum oils was observed.

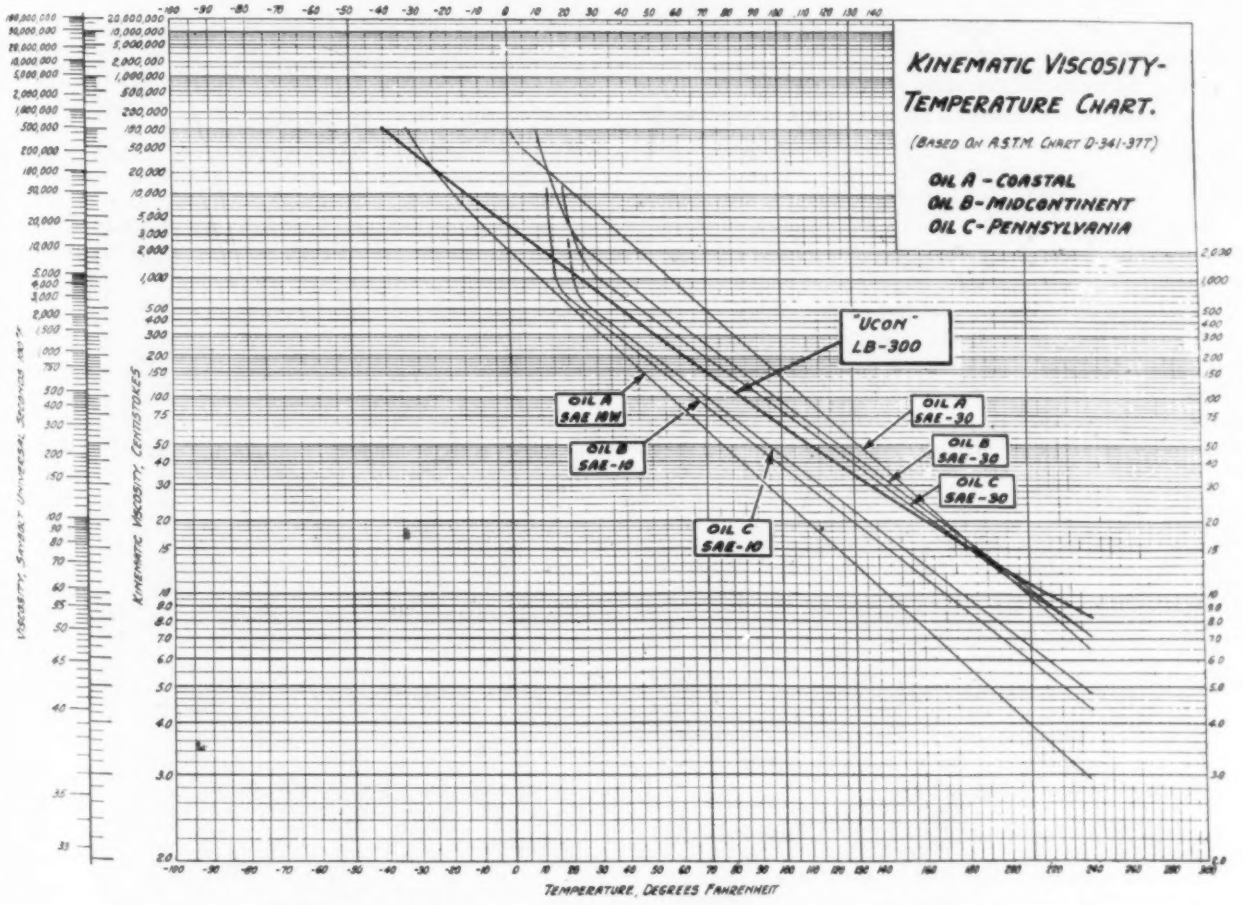
(h) It was established that engines operated on the synthetic lubricant were much easier to clean up during overhaul than engines operated on petroleum oil. Servicing aircraft was much easier in cold weather and aircraft could be left out of doors or uncovered for longer periods during loading and unloading.

Other Uses for LB Series

In addition to use in internal-combustion engines there are many places where the special properties of the LB series of synthetic lubricants, such as low pour point, high viscosity index, and nonsludging characteristics, can be utilized to advantage. In addition to their use as hydraulic fluids and plasticizers, utilization as lubricants in many fields is being studied. These include powdered metal bearings, transmission and differential lubricants, electric motors, textile machinery, wire drawing, metal rolling and stamping, and many others. Greases exhibiting unusual high- and low-temperature properties have been prepared from oils of the LB series, so far on an experimental basis only.

50-HB Series

So far this paper has been confined to discussion of the LB (water-insoluble) series of synthetic lubricants. Physical data for the 50-HB (water-soluble) series are given in Table 10. This series is characterized by high viscosity index, low pour point, and in other properties is similar to the LB series. It is the equal of the LB series in lubricating ability. Compounds of this series are completely soluble in water at room temperatures, the water solubility decreasing as the temperature is raised. The 50-HB series will depress the freezing point of water to a limited extent. Members of the 50-HB series having viscosities of more than about 100 SUS at 100 F have very little swelling action on natural rubber and are excellent lubricants for both rubber and metal. Certain of these products are being used in hydraulic fluids including brake fluids, as high-temperature heat-transfer media, textile lubricants, in cutting oils, for wire and metal drawing operations, and other uses.



■ Fig. 13 – Viscosity-temperature relationships – petroleum oils and synthetic

ELECTRICAL MODEL for Investigation of

by HUGH B. STEWART
Allison Division
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INCREASED power in aircraft engines has focused attention on the torsional vibration of the crankshaft system in in-line engines. Dangerous resonances in the operating range of the engine often lead to failures in engine parts or the propeller. Thus, it is important to obtain preliminary information regarding the vibration characteristics of new engine designs and to be able to predict how various changes in design will affect the torsional vibration.

The natural frequencies and the relative amplitudes of vibration for the crankshaft system are usually determined by the Holzer tabulation method. The mathematics involved in the method is not difficult; however, since it is an iteration process, it is usually necessary to make several trials for each mode of vibration before the natural frequency is determined. The solution for the first three modes of vibration for an in-line engine often requires several days' work. If it is desired to determine the effect of changing the rigidity or inertia of some part, for example, a pendulum damper, on the natural frequencies of the engine, the calculations must be repeated for each of the new values. Because of the time required to make a complete mathematical analysis of an engine system, the solution in many cases becomes impractical.

If the harmonic coefficients of the torque curve are known, the Holzer table can be modified to calculate the forced vibrations. However, this also requires considerable time and has the further disadvantage that at resonance frequencies the amplitudes become infinite. If the friction in the system can be evaluated it is mathematically possible to compute the actual tuning curve, but the calculation is very complicated.

This paper describes an electrical model which has been developed at the Allison Division of General Motors Corp. to simplify the study of torsional vibration of in-line engines. In a relatively short time it is possible with this model to determine the natural frequencies and relative amplitudes of vibration for an engine system. The effect of changing the rigidity or inertia of various elements can be easily and quickly investigated. By adding a damping factor to the system the absolute amplitudes of vibration can be determined with the model. Provisions are made to study the effect of various firing orders on the vibration characteristics. The torque in any part of the crankshaft or to the propeller can be found, and the effectiveness of different pendulum damper tunings as well as the pendulum bob amplitudes can be determined.

The electrical model is based on certain fundamental analogies between mechanical and electrical oscillating systems. The analogy is most apparent for the case of an electrical circuit consisting of a condenser and an inductance and a mechanical system consisting of an inertia and a rigidity element. If certain relations are established between the electrical and mechanical units, the resonance characteristics of the two systems will be similar. In fact, it is possible to determine completely the vibration characteristics of the mechanical system by measuring the voltages across the elements of the electrical circuit.

Two methods for representing mechanical systems by electrical circuits have been developed. A detailed discussion of these methods can be obtained by consulting the references.[1, 2] A brief review of the principles is presented here.

Consider the mechanical system illustrated in Fig. 1*A*, which consists of an inertia I on a shaft of torsional stiffness K and with a damping constant b. If a torque T is applied to the inertia the equation of motion is:

$$I\frac{d^2\theta}{dt^2} + b\frac{d\theta}{dt} + K\theta = T$$

or writing $\Omega = \frac{d\theta}{dt}$, where Ω is the angular velocity of torsional vibration:

$$I\frac{d\Omega}{dt} + b\,\Omega + K\int \Omega\, dt = T \qquad (1)$$

Fig. 1*B* shows an electrical circuit consisting of an inductance, resistance, and capacity in series. Since the voltages add in a series circuit, the differential equation is:

$$L\frac{di}{dt} + Ri + \frac{1}{C}\int i dt = e \qquad (2)$$

where e is the applied voltage.

If the elements are in parallel as in Fig. 1*C*, the currents through the individual elements add, and the differential equation is:

$$C\frac{de}{dt} + \frac{1}{R}e + \frac{1}{L}\int e dt = \qquad (3)$$

where i is the supplied current.

[This paper was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Nov. 1, 1945.]

[1] See *Journal of Applied Physics*, Vol. 9, June, 1938, pp. 373-387: "Mobility Method of Computing Vibration of Linear Mechanical and Acoustical Systems: Mechanical-Electrical Analogies," by F. A. Firestone.

[2] See *Journal of Acoustical Society of America*, Vol. 14, January, 1943, pp. 183-192: "Applications and Limitations of Mechanical-Electrical Analogies, New and Old," by J. Miles.

AN electrical analyzer is described here which was designed in order to solve crankshaft torsional vibration problems.

THE AUTHOR: HUGH B. STEWART has been with Allison Division, GMC, as test engineer in the field of vibration and stress analysis since 1941. Prior to that Mr. Stewart, an alumnus of Kent State University, was taking graduate work at Ohio State University.

CRANKSHAFT TORSIONAL VIBRATIONS In In-Line Engines

It is apparent that either equation (2) or (3) can be selected as the electrical analogue of equation (1). In the older mechanical-electrical analogy, torque was associated with voltage and velocity with current. Thus the electrical circuit corresponding to the mechanical system was series connected. In the newer method of analogy, which Firestone has designated the mobility method, torque is associated with current and velocity with voltage, in which case the electrical circuit is parallel connected. By referring to equations (1), (2), and (3), the mechanical-electrical associations are seen to be as follows:

Mechanical	Electrical	
	Old	New
I	L	C
b	R	$1/R$
K	$1/C$	$1/L$
T	e	i
Ω	i	e

If the series circuit is chosen, the voltage *across* an electrical element is associated with the torque *through* a mechanical element. Similarly, current *through* an electrical element corresponds to velocity *across* a mechanical element. This fact is a source of confusion in representing mechanical systems by electrical circuits, particularly when the systems become complicated. The mobility method eliminates this difficulty by associating voltage *across* with velocity *across* and current *through* with torque *through*. The latter method was adopted for the electrical model since it simplified the circuit representation for the more complex engine designs. An additional advantage of the mobility model is that velocity resonances can be measured more easily by voltages than by currents.

The following rules of mechanical circuit connections have been adopted from the mobility method:

1. In series connections the terminals of the elements are connected end to end with no more than two terminals to any junction point.

2. In parallel connections the terminals of the elements are connected to common junction points.

3. For a mass element, one terminal is on the mass itself while the other terminal is the fixed frame or ground.

Having determined the mechanical circuit for the system, the analogous electrical circuit is apparent.

Since voltage is associated with velocity in the mobility method, a voltage resonance in the electrical circuit indicates a velocity resonance in the mechanical system.

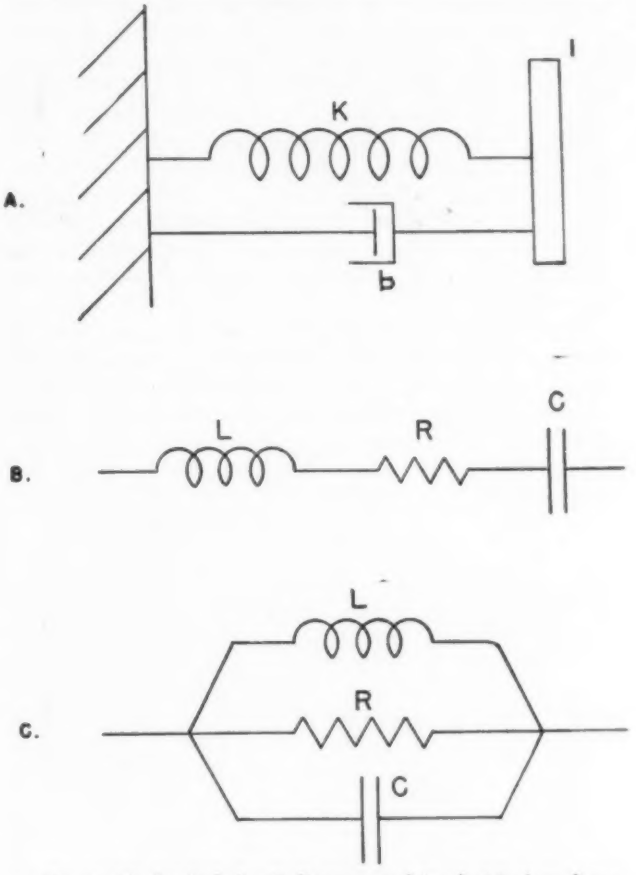
If a mechanical circuit contains no friction, the frequencies of the velocity resonance, displacement resonance, and the natural frequency are all the same. However, when friction is added to the mechanical circuit, these frequencies will in general be changed from their original values and will differ from each other. Thus, it is possible that the displacement resonance frequency for a system including friction may be appreciably different from the resonance frequency of the system without damping. If the friction elements are included in the circuits, as well as the inertia and rigidity elements, an absolute value of the velocity amplitude can be obtained for a particular torque amplitude. Knowing the frequency of vibration, the displacement amplitude can be determined from the equation

$$\Omega = j\,\omega\,\theta$$

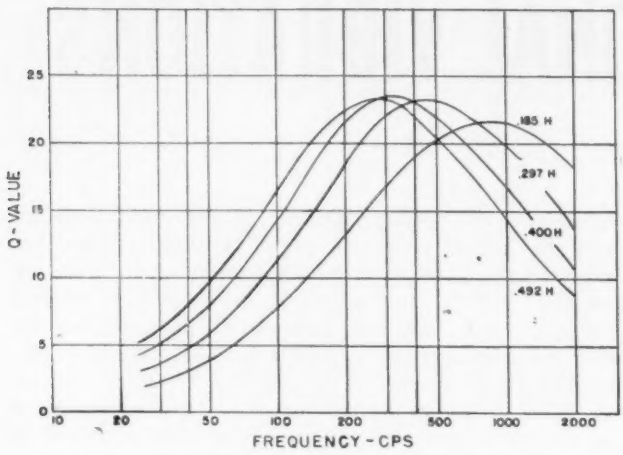
where the j indicates Ω is 90 deg out of phase with θ, and:

$$|\,\theta\,| = \frac{\Omega}{\omega} \qquad (4)$$

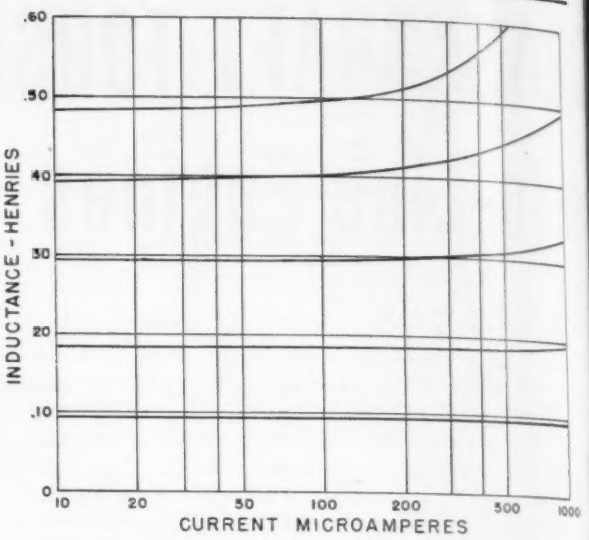
The complete evaluation of mechanical friction elements in the engine is a difficult problem. The representation of these elements in the electrical model is further compli-



■ Fig. 1 – Mechanical circuit diagram and its electrical analogues



■ Fig. 2 – Effect of frequency and air gap size on dissipation factor for coil with iron core



■ Fig. 3 – Effect of current and air gap size on inductance of coil with iron core

cated by the fact that a large part of the friction forces in the engine is not proportional to velocity. Calculations and torsional vibration measurements on the Allison engine show that the most important source of friction for the first two modes of vibration is the hydraulic damper unit which is employed in the engine to dissipate vibration energy. Tests conducted on a damper calibrating machine indicate that the damping torque of the unit can be considered proportional to the angular velocity for a limited range of frequencies. Introducing the damping coefficient, as determined from the calibrator, into the electrical model, it was found that satisfactory agreement with engine test results could be obtained for the first two modes of vibration although other sources of friction were ignored. This procedure, however, is not justified in the case of the third mode of vibration where the bearing and piston friction and the hysteresis damping seem to be the two predominant sources of friction. Because of the difficulty in evaluating and representing these factors in an electrical model, the effect of damping in this mode of vibration has not been studied.

In the theory of the mechanical-electrical analogies it was assumed that the electrical elements were pure inductances, capacities, and resistances. Actually, this is not the case, since the inductances and capacities both contain dissipation factors which alter the theory somewhat. The dissipation of the condensers is usually negligible in comparison to that of the inductances. The resistance of the coils cannot be compared to friction losses in the engine since the coil resistance is effectively in series with the inductance, while the friction resistance should be in parallel. Using vector notation, the impedance of the coil is

$$Z = R + jL\omega$$

where R is the resistance of the coil, and the admittance is:

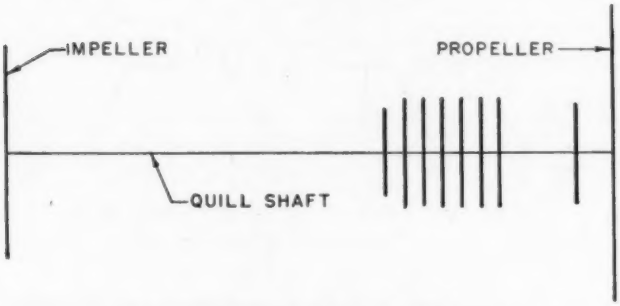
$$Y = \frac{1}{R + jL\omega} = \frac{R}{R^2 + L^2\omega^2} - \frac{jL\omega}{R^2 + L^2\omega^2}$$

The susceptance of the coil as measured by an impedance bridge is $-j/L\omega$ while the susceptance as seen in the parallel circuit is:

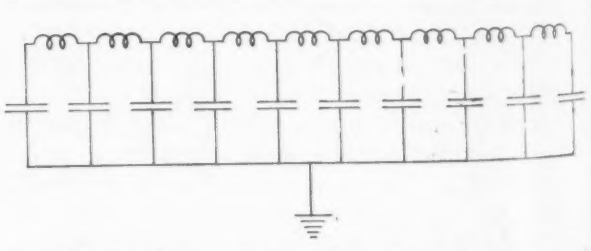
$$B = -\frac{jL\omega}{R^2 + L^2\omega^2}$$

Thus, for large R the susceptance will be low, which will cause the circuit to tune low.

In addition to causing the circuit to tune low the coil resistance will also make the tuning curve broad. In order to keep the losses of the coils low, they were all layer wound with low-resistance wire and a good grade of iron was chosen for the core material. An adjustable air gap in the iron core provided a means for varying the inductance of the coils over small ranges. Fig. 2 shows a graph of the Q-value, that is, the ratio $L\omega/R$, for a sample coil as a function of the frequency for various air gaps in the iron core. These curves were obtained from impedance bridge measurements with a current of approximately 200



■ Fig. 4 – Inertia distribution of 12-cyl engine



■ Fig. 5 – Electrical circuit for mechanical system shown in Fig. 4

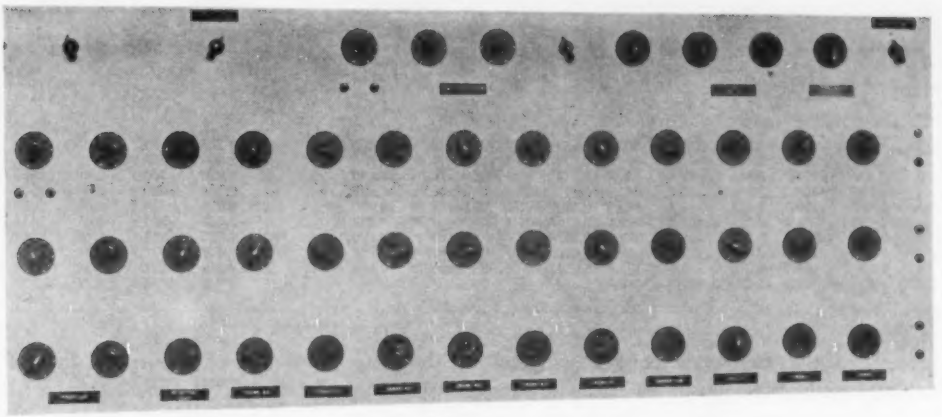


Fig. 6 – Panel view of electrical model

microamp through the coil. This particular coil had 740 turns of No. 26 wire and corresponded to the coils used for the crankshaft rigidities. The curves indicate that the best Q-values can be obtained at the low frequencies by making the air gap in the core small.

Although the Q-value of the coils improved for smaller air gaps, it was found that the smaller gaps caused the inductance to change slightly with increasing currents. This effect is illustrated by Fig. 3 for the same coil. Since the latter effect was the more serious of the two, it was necessary to use a fairly large air gap and limit the current flow through the inductances in the electrical model.

No appreciable change in Q-value could be detected for this coil when the frequency and inductance were held constant and the current was varied within the prescribed limits.

Equation (1) can be multiplied through by some constant m without altering the equality. This provides a method of adjusting the coefficients to values which can be conveniently represented by electrical constants. The values of C, R, and L chosen for the model were

$$C = mI,\ R = 1/mb,\ L = 1/mK,$$

where $m = 0.500 \times 10^{-4}$. This makes $i = mT$, provided that $e = \Omega$.

However, the value of Ω can also be multiplied by some constant n, which changes T by a corresponding amount.

Thus if:

$$e = n\Omega \tag{5}$$

then

$$i = mnT \tag{6}$$

So there is some freedom in choosing the current and voltage as well as C, R, and L in the electrical circuit. The value chosen for n was such that the current through the coil remained sufficiently low so that no appreciable inductance changes occurred with varying currents, as discussed previously.

It is noted that the unit of time can also be multiplied by some constant which will consequently change the values of the inductance and capacity. For example, going to a higher frequency would make it possible to decrease the size of the inductance and capacity. However, when the frequency is increased appreciably, the Q-value of an iron core coil decreases. Since the iron core and variable air gap provide such a convenient method for adjusting the inductances, it was preferred to work at the lower frequencies. Thus, time in the electrical model was chosen to have the same magnitude as in the engine system. Fig. 4 shows the mechanical system for a 12-cyl Allison engine, while Fig. 5 is the corresponding circuit diagram of the electrical model.

Fig. 6 shows the panel of the electrical model and Fig. 7 is a rear view showing the condenser banks and the inductances. Decade condensers at each of the crankshaft positions make it possible to adjust the inertias to a 1.0% accuracy. A bank of condensers representing the pendulum damper is provided with a selector switch so that it can be applied at any position in the crankshaft system. The circuit of the model is arranged so that the exciting current can be applied to any point in the system by means of a selector switch. Two other selector switches and outlets provide a means for measuring the voltage amplitude at any place in the circuit, or the voltage at any two points can be recorded on an oscillograph simultaneously by means of an electronic switch. The latter enables one to determine the phase relation between any two points along the crankshaft and thus locate nodes. A vacuum tube voltmeter is used to measure the voltage amplitudes.

It was desired to have a constant torque input at different frequencies for the model, which required a constant current supply. It was found that the impedance of the model circuit never exceeded 20,000 ohms, thus by putting a 2.0 megohm limiting resistance in series with the circuit the current input could be kept constant to within 1%

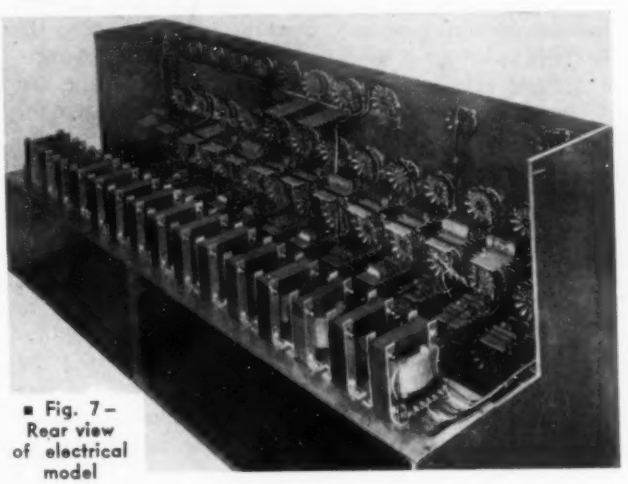


Fig. 7 – Rear view of electrical model

Table 1 – Calculations for $f = 27.4$ cps and $\omega = 172.1$

Mass No.	C	$-C\omega$	e	$i=\frac{e}{X_C}$	i	L	$L\omega$	$e_d=iX_L$
1	5.245	-902.7	1.000	-902.7	-902.7	5.319×10^{-6}	915.4×10^{-6}	-0.826
2	0.400	-68.8	0.174	-12.0	-914.7	0.0714×10^{-6}	12.29×10^{-6}	-0.011
3	0.620	-106.7	0.163	-17.4	-932.1	0.1176×10^{-6}	20.24×10^{-6}	-0.019
4	0.620	-106.7	0.144	-15.4	-947.5	0.1176×10^{-6}	20.24×10^{-6}	-0.019
5	0.620	-106.7	0.125	-13.3	-960.8	0.1205×10^{-6}	20.74×10^{-6}	-0.020
6	0.620	-106.7	0.105	-11.2	-972.0	0.1176×10^{-6}	20.24×10^{-6}	-0.020
7	0.620	-106.7	0.085	-9.1	-981.1	0.1176×10^{-6}	20.24×10^{-6}	-0.020
8	0.649	-111.7	0.065	-7.3	-988.4	0.4425×10^{-6}	76.15×10^{-6}	-0.075
9	0.518	-89.1	-0.010	0.9	-987.5	0.2381×10^{-6}	40.98×10^{-6}	-0.040
10	114.0	-19,620.0	-0.050	981.0	-6.5			

Table 2 – Calculations for $f = 92.5$ cps and $\omega = 581.2$

Mass No.	C	$-C\omega$	e	$i=\frac{e}{X_C}$	i	L	$L\omega$	$e_d=iX_L$	
1	5.245	-3048	1.00	-3048	-3048	5.319×10^{-6}	30.91×10^{-6}	-0.42	
2	0.400	-232.5	-8.42	1958	-1090	0.0714×10^{-6}	41.50×10^{-6}	-0.08	
3	0.620	-360.3	-8.47	3052	1962	0.1176×10^{-6}	68.35×10^{-6}	0.13	
4	0.620	-360.3	-8.34	3005	4967	0.1176×10^{-6}	68.35×10^{-6}	0.34	
5	0.620	-360.3	-8.00	2882	7849	0.1205×10^{-6}	70.03×10^{-6}	0.53	
6	0.620	-360.3	-7.45	2684	10,530	0.1176×10^{-6}	68.35×10^{-6}	0.72	
7	0.620	-360.3	-6.73	2425	12,960	0.1176×10^{-6}	68.35×10^{-6}	0.89	
8	0.649	-377.2	-5.84	2203	15,160	0.4425×10^{-6}	257.2×10^{-6}	2.58	
9	0.518	-301.1	-1.94	504	15,740	0.2381×10^{-6}	138.4×10^{-6}	2.11	
10	114.0	-66,260	0.24	-15,900	-160				

throughout the frequency range, provided the oscillator current supply was constant. Allowing for variations in the oscillator output voltage, the current probably does not vary over 3% in the important range of frequencies.

The application of the electrical model to the investigation of the torsional vibration characteristics of a particular Allison engine is described below. In order to indicate the accuracy of the model, a mathematical analysis of this engine is included for comparison.

The procedure which has been used in the calculation of the natural frequencies and relative amplitudes of the system is the electrical analogue of the Holzer method. This method has no particular advantage over the Holzer method, but has been used to demonstrate further the analogy between electrical and mechanical vibrating systems. A brief description of the method is presented here. A more detailed analysis can be formulated by applying Kirchhoff's laws to the electrical circuit.

The results of the calculations for this engine are tabulated for the first three modes of vibration. (See Tables 1, 2, and 3.) The values of the inertias are placed in the capacity column and the flexibilities in the inductance column. A frequency is estimated and the capacitive and inductive reactances are calculated for this frequency. The voltage drop across the first capacity is arbitrarily assigned a value of 1.0 v. The current through the condenser is calculated for this potential drop. Since all of this current must flow through the inductance, the voltage drop across the inductance can be calculated. The voltage across the second capacity is equal to the original voltage plus the voltage drop across the inductance. From this the current flow through the second capacity can be calculated. The current flow in the second inductance is the sum of the current through the first one and the current which passed through the condenser, the phase being taken into account. This procedure is continued until the current through the last inductance and the voltage drop across the last condenser are calculated. The difference between the last inductance current and the last condenser current is the current which must be supplied to the circuit at this point to maintain the specified voltages across the various elements.

The condition of voltage resonance means that no external current supply is necessary to maintain finite alternating voltages across the elements, provided, of course, there is no dissipation in the network. Then, in order to obtain voltage resonance in the circuit discussed, the frequency must be adjusted until the residual current in the calculation is zero. The voltages at the junction points represent relative velocities along the crankshaft system, and for a given frequency correspond to relative amplitudes.

Fig. 8 illustrates the voltage resonance curve of the model for the case where constant current input is supplied to the rear of the crankshaft and the voltage output is measured at the same position. Fig. 9 shows the relative vibration amplitudes along the crankshaft system as obtained from the electrical circuit. Since the resonance frequencies and the relative amplitudes of the system at resonance are independent of the position of current application, any convenient position can be chosen to apply the current to determine these values. Table 4 shows the resonance frequencies obtained from the electrical model as compared to the calculated values. The second and third are the important modes of vibration in the engine. The frequency accuracy of the electrical model in the important range of frequencies is found to be approximately 1%. For measuring amplitudes it is found that the mean deviation of the measured values from the calculated values is less than 2%, while the maximum deviation is about 5%.

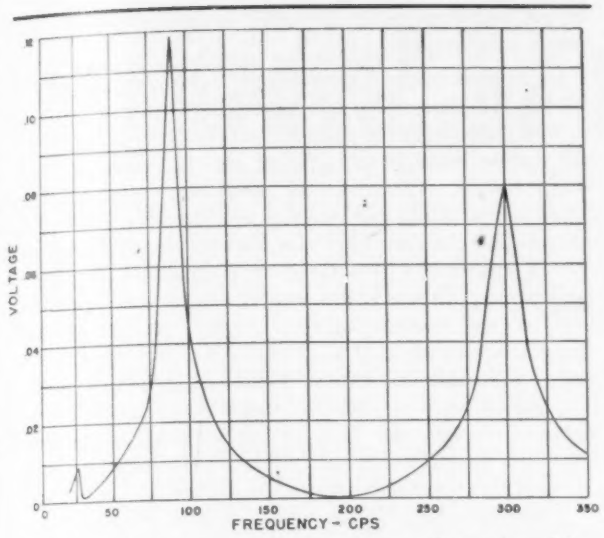
By applying a potential of 20 v across the limiting resistance of 2 megohms, the maximum current through the inductances in the electrical circuit was limited to approximately 100 microamp. The current calibration curves for the coil, Fig. 3, indicate that the variation in inductance is negligible for currents below 100 μA while the inductance is about 3% high for a current of 400 microamp. Table 5 is a comparison of the resonance frequencies of the circuit for various input currents. It is seen, for example, that the second mode resonance frequency is about 3% low when the current input is such that the maximum current through the inductances approaches 400 microamp, that is, for an input of 40 microamp.

A hydraulic damper is used in the Allison engine to dissipate the energy in the second mode resonance. The friction is applied between the rear of the crankshaft and the supercharger impeller. The optimum value of damping can be determined by means of the electrical model.

Table 3 – Calculations for $f = 301$ cps and $\omega = 1892$

Mass No.	C	$-C\omega$	e	$i=\frac{e}{X_C}$	i	L	$L\omega$	$e_d=iX_L$
1	5.425	-9929	1.00	-9929	-9929	5.319×10^{-6}	$10,070\times10^{-6}$	-108.0
2	0.400	-757.2	-99.0	74,960	65,030	0.0714×10^{-6}	135.2×10^{-6}	8.8
3	0.620	-1174	-90.2	105,900	170,900	0.1176×10^{-6}	222.6×10^{-6}	-38.0
4	0.620	-1174	-52.2	61,300	232,200	0.1176×10^{-6}	222.6×10^{-6}	51.7
5	0.620	-1174	-0.5	590	232,800	0.1205×10^{-6}	228.1×10^{-6}	53.1
6	0.620	-1174	52.6	-61,800	171,000	0.1176×10^{-6}	222.6×10^{-6}	38.1
7	0.620	-1174	90.7	-106,500	64,500	0.1176×10^{-6}	222.6×10^{-6}	14.4
8	0.649	-1229	105.1	-129,200	-64,700	0.4425×10^{-6}	837.6×10^{-6}	-54.2
9	0.518	-980.6	50.9	-49,900	-114,600	0.2381×10^{-6}	450.7×10^{-6}	-51.6
10	114.0	-215,800	-0.7	151,100	36,500			

$$X_C = -\frac{1}{C\omega} \qquad X_L = L\omega$$

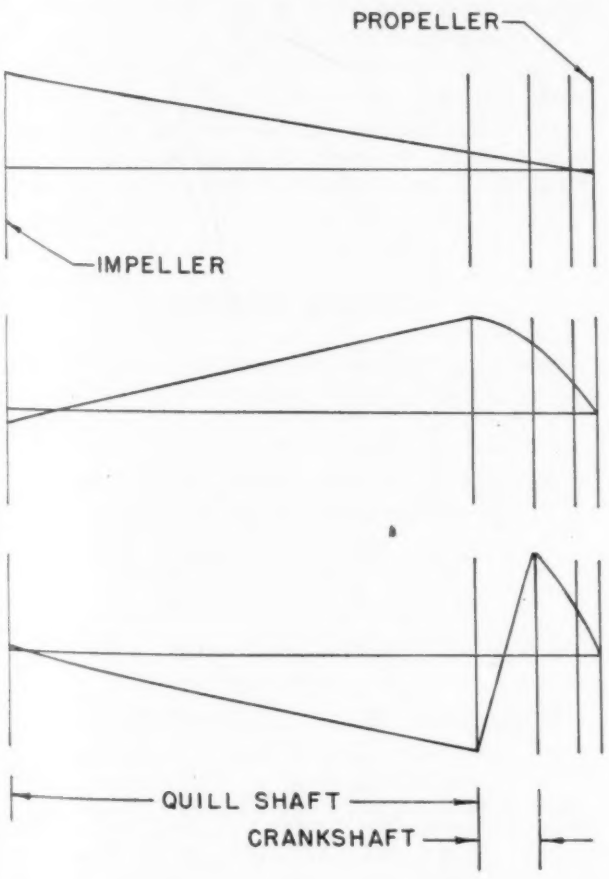


■ Fig. 8 – Resonance curve obtained from electrical model

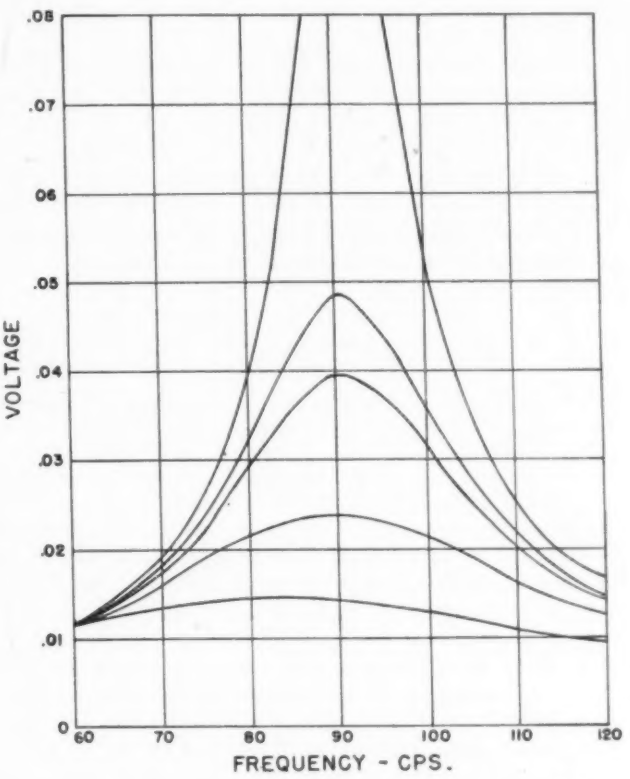
Hydraulic dampers having different characteristics have been calibrated on a special machine which makes it possible to assign a particular damping coefficient to each damper. The appropriate resistance corresponding to the damping coefficient is then shunted across the quill shaft inductance. Fig. 10 illustrates some voltage resonance curves measured at the rear of the crankshaft when damping coefficients of $b = 214$, 294, 588, and 1176 lb-in.-sec were applied at the hydraulic damper position.

In some cases it has been necessary to study the forced vibration characteristics. For these cases it is desirable to have a current generator which will apply the current simultaneously to all six crankthrows with the same phase relation as in the engine. The method which was adopted corresponds to the analytical method. Fig. 11 illustrates the generator circuit which is used to study the particular minor orders 1½, 4½, 7½, and so on, when the engine firing order is 1-5-3-6-2-4. For these orders the successive impulses at the crankthrows are 180 deg apart. In the case of the sixth order, which is a major order, the phase angle between the impulses at the different crankthrows is 360 deg and the points A and B must both be connected to the same side of ground, that is, point D. Among the engine designs which have been tested on the electrical model to date, it has only been necessary to study the forced vibration for the minor orders mentioned above. Other minor orders could be studied by using condenser dividers in place of the $C's$ in the example, and by making two separate determinations for the sine and cosine components of the impulses.

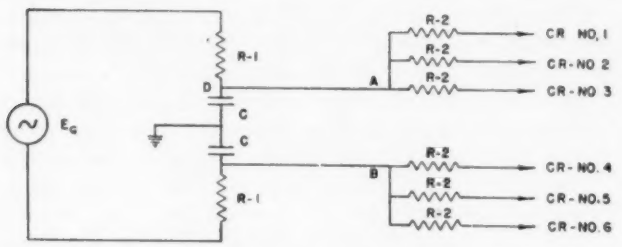
The resistances R_1 are large in comparison to the reactance of C at the lowest operating frequencies. If the voltage of the oscillator is constant the voltage drop across the condensers is inversely proportional to the frequency, and the voltage output of the model is then proportional to the torsional vibration displacement instead of velocity. The



■ Fig. 9 – Relative amplitudes along crankshaft system for first three modes of vibration



■ Fig. 10 – Resonance curves for different damping coefficients across quill shaft



■ Fig. 11 – Current generator circuit for applying firing torques

resistances R_2 isolate the crankthrows from each other and from the condenser C.

Since the resistance of R_1 is large in comparison to the reactance of C, the current through C is approximately:

$$i_1 = \frac{e_g}{Z} = \frac{e_g}{2R_1}$$

and the voltage drop across C is:

$$e_C = i_1 X_C = \frac{e_g}{2 R_1 C\omega}$$

The rms current supplied to the model through each of the resistances R_2 is then:

$$i_2 = \frac{e_C}{R_2} = \frac{e_g}{2R_1 R_2 C\omega} \tag{7}$$

where R_2 is large in comparison to the impedance of the model. The relation between the rms current and the torque amplitude for the model is from equation (6):

$$i_2 = \frac{mn T_o}{\sqrt{2}}$$

so

$$\frac{m n T_o}{\sqrt{2}} = \frac{e_g}{2R_1 R_2 C\omega} \tag{8}$$

Also from equations (5) and (4) the relation between the rms voltage and the displacement amplitude is:

$$e = \frac{n \Omega_o}{\sqrt{2}\ 57.3} = \frac{n}{\sqrt{2}} \omega \frac{\theta_o}{57.3} \tag{9}$$

where θ_o is the amplitude in deg. Eliminating n between equations (8) and (9) and solving for θ_o:

$$\theta_o = \frac{(114.6\ m\ R_1\ R_2\ C)}{e_g} T_o\ e \tag{10}$$

Thus for a constant oscillator voltage e_g the torsional vibration amplitude at any point can be obtained if the voltage at that point is measured. The torque T_o is the torque amplitude at each crankthrow for the particular order being investigated.

The torque transmitted through any section of the crankshaft system can be determined by measuring the voltage drop across this section. Then since:

$$T = K\ \theta \tag{11}$$

the torque can be calculated from equations (10) and (11), except that θ must be converted to radians before substituting into equation (11).

Using the torque generator the actual resonance curve for the engine system can be determined if the correct damping coefficient is applied to the model. Furthermore, the torque through any part of the crankshaft system is easily found. The effect of pendulum dampers on the torsional vibrations can easily be studied with the electrical model. For the case where the damper is tuned to infinite inertia, the circuit is shorted to ground at the point of application. This provides an effective means of determining the best location for the damper. The effect of the damper on other orders or the effect of inertia tuning (other than infinite) can be studied by applying an effective inertia to the system as calculated from the tuning curve of the damper. If the effective inertia is positive, a condenser is applied to the circuit while a negative inertia requires the application of an inductance.

In the particular example discussed in this article, the propeller has been considered a pure inertia element. Actually, the propeller has its own modes of vibration and when combined with the crankshaft system will, in general, shift the resonance frequencies and create new resonances. By experimentally determining the effective inertia or dynamic modulus of the propeller, the inertia of the propeller can be corrected in the model to the effective inertia. In practical cases it has been found that the propeller can usually be considered as a pure inertia, particularly in engines incorporating flexible extension shafts between the crankshaft and the propeller.

In general, good agreement between engine results and the electrical model has been experienced, thus making the model a valuable instrument.

The author wishes to acknowledge the helpful suggestions of J. M. Whitmore, head of the Electronics and Vibration Group, and the technical assistance of P. W. Nelson.

Additional Reference

"Mechanical Vibrations," by J. P. Den Hartog. McGraw-Hill, New York, 1934, pp. 225-233.

Table 4 – Resonance Frequencies Obtained from Electrical Model as Compared with Calculated Values

Mode	Calculated Frequency, cps	Model Frequency, cps
1	27.4	26.3
2	92.5	91.5
3	301	300

Table 5 – Comparison of Resonance Frequencies of Circuit for Various Input Currents

Input Current, Microamp	Resonance Frequency, cps			Error, %		
	First Mode	Second Mode	Third Mode	First Mode	Second Mode	Third Mode
5	26.7	91.5	300	2.6	1.1	0.3
10	26.3	91.5	300	4.0	1.1	0.3
20	25.0	91.0	299	8.8	1.6	0.7
40	23.6	90.0	299	13.9	2.7	0.7
100	22.2	86.5	296	19.0	6.5	1.7
200	21.7	85.0	292	20.8	8.1	3.9

ENGINEERING of INVOLUTE SPLINES

by GEORGE L. McCAIN
Staff Engineer – Research
Chrysler Corp.

Written discussion of the paper by Mr. McCain will be published in full in the June issue of the SAE Journal (Transactions).

WRITTEN shortly before the adoption of the SAE standard on involute splines, Mr. McCain presents in this paper the engineering basis on which the standard was formulated, setting forth the reasons why it should have widespread use in the hands of automotive and general mechanical equipment manufacturers, and the suppliers of tools, gages, and so on.

In addition, Mr. McCain gives the historical background of the project as a cooperative accomplishment with other groups, such as the national associations of the general mechanical, gear and tool industries.

THE AUTHOR: GEORGE L. McCAIN (M '12), who has been in the automotive engineering field since his graduation from Iowa State College in 1909, is now concerned with research for Chrysler Corp. Previously on the staffs of the Packard, Lincoln and Dodge companies, Mr. McCain joined Chrysler in 1928.

STANDARDIZATION of involute splines goes back originally to 1930, when Sectional Committee B5 (sponsored by the Society of Automotive Engineers, American Society of Mechanical Engineers, and the National Machine Tool Builders Association under the American Standards Association) organized Technical Committee 13, which developed a side bearing involute spline standard that was approved in 1939 as American Standard B5.15-1939. These splines were a flat root type, based on metric shaft diameters from 20 to 110 mm, corresponding to ball-bearing bores.

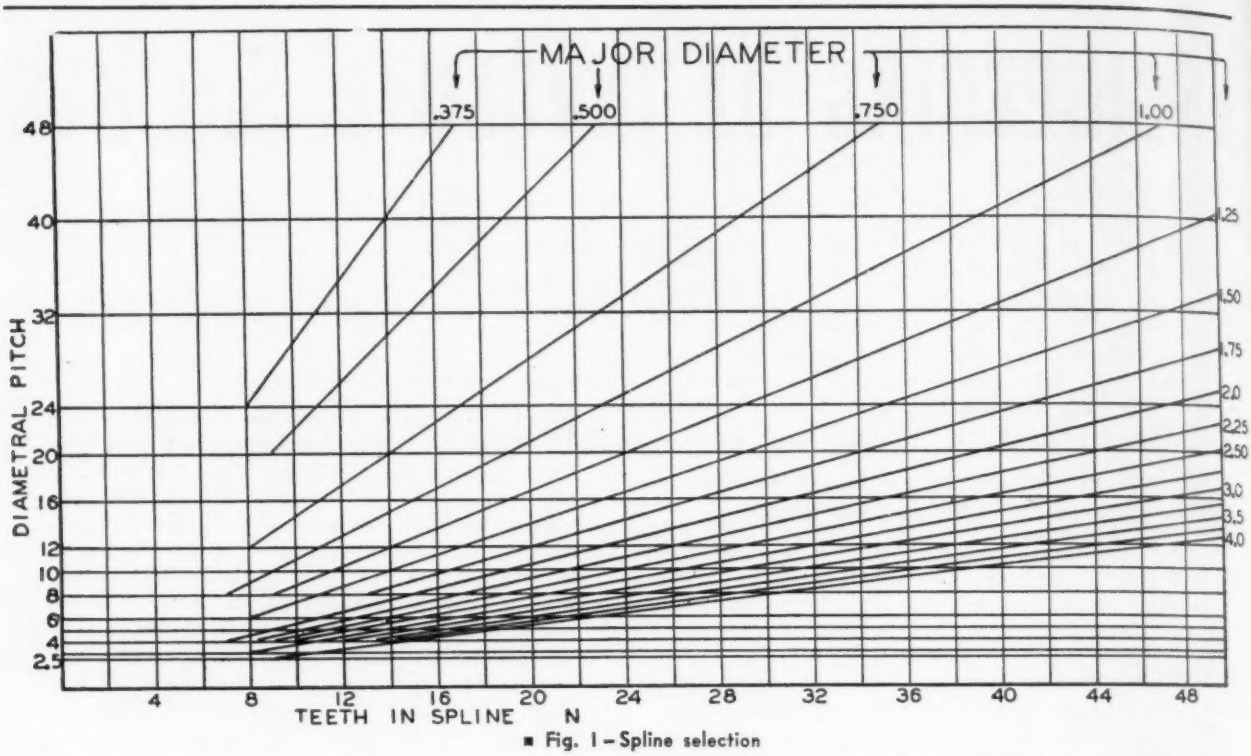
As this standard was not acceptable to the aircraft industry, the SAE Aeronautics Division developed SAE Aeronautic Standard AS-84, which was adopted in 1942. This is a full fillet root type ranging from 6/12 to 48/96 diametral pitches.

In general practice, need was evidenced for a standard covering inch diameters as well as metric, and accordingly the SAE Parts and Fittings Division, the Spline Committee of the American Gear Manufacturers Association, and Technical Committee 13 of ASA Sectional Committee B5 coordinated their efforts toward setting up a common standard for general practice. The SAE Aeronautics Committee E-7 also participated in view of the intent at that time to extend AS-84 downward to include diametral pitches to ½ for adoption as the ASA standard, but it was found that this full fillet type of spline was too deep for many applications. Accordingly, a flat root type of spline was developed, the first version of which was issued in October, 1944, in pamphlet form. The committees under ASA recognized the desirability of making the splines of the new standard interchangeable with AS-84, but the small fillet formed at the root of the tooth by generating processes requires a slight chamfer at the tip of the mating tooth, which was not incorporated in AS-84. In the new standard, a full fillet type is included, in which the necessary modification of major and minor diameters ensures interchangeability with the flat root type.

The problem of fastening together two machined parts to prevent relative rotation has been solved in several ways, depending on conditions. The use of a key, or multiple keys, has been general practice for generations, but one great problem was the angular spacing of the keyways so that all parts would work together. Some parts were put together permanently, and others were made to slide, loaded or unloaded. Some of us can remember multiple disc clutches as first used on automobiles. One method was to drive the discs by pins set into the flywheel. The discs were tried in several positions and usually went on to the pins in only one place. This resulted in point bearings at only a few places, which soon wore loose and rattled. Then along came a shallow straight spline in a shell bolted to the flywheel. Spacing was a problem still, and uniform bearing was accidental. The next development was an involute form of spline which could be generated on both internal and external members. This solved the spacing problem and resulted in better bearing area.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1946.]

The equations and their identifying numbers, as well as Appendix I, Tables IV, XI, XX, XXI, and XXVII, mentioned in the paper, refer to equivalent items in the SAE standard on involute splines as it will be published in the SAE Handbook. Likewise, the numbers placed in brackets at the end of some of the captions are the figure numbers these same drawings will have in the standard.



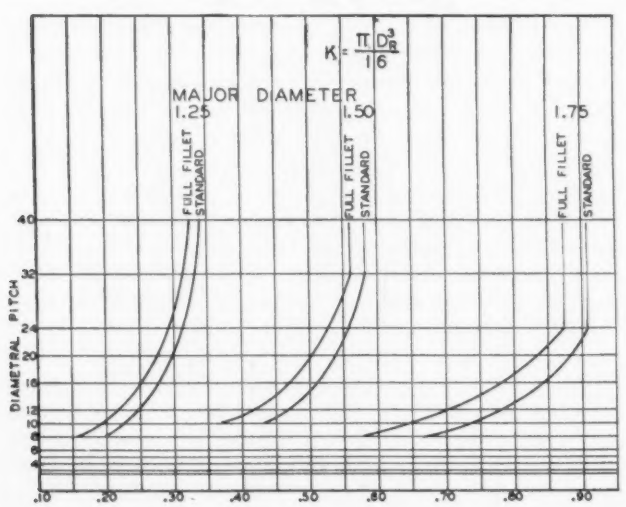
■ Fig. 1 – Spline selection

There are other examples. The clutch shaft coupling with the transmission main shaft was first made by drilling a circle of holes, then machining away the inner half, and using the resulting half-holes to engage teeth on the sliding gear. These teeth could be indexed, but the holes could not be spaced accurately. The gear would creep out under load and become loose. When these parts were made with involute-form teeth, a big improvement resulted.

In 1920 Douglas T. Hamilton[1] reviewed current practices, and here I quote:

"The multiple spline has obvious advantages over the single key, but it will be of interest to mention briefly wherein an increased number of splines is superior to a smaller number. The square shaft, which was the progenitor of the splined shaft, has several disadvantages. It is difficult to obtain a good fit between the two parts, absolute concentricity is almost impossible to attain, and it has a much greater tendency to burst the shell. The use of involute curves for the sides of splines, instead of having them straight-sided, is a still further improvement. There are several reasons why the involute-sided spline is better than one having straight sides. The involute spline can be generated with a tool which does not change its shape when resharpened; hence, greater accuracy, with its attendant possibilities for interchangeability, can be attained. Further, due to the curved inclination of the sides of the spline, especially on a splined shaft upon which sliding gears are held, there is a tendency for the gears always to remain concentric with the shaft when under load. This is not true to the same extent with the straight-sided spline; consequently, much closer fits between diameters are necessary than is the case with an involute spline. It is also a well-known fact that a generating tool used for cutting gear teeth does not necessarily have the same shape as the tooth it produces. Further, with the generating principle, the tooth thickness can be varied without changing the involute shape, simply by feeding the generating tool in farther or holding it back. This flexibility is not possible with the forming process. The involute spline can be generated just as accurately as a gear tooth and, as will be explained, has other advantages."

[1] See *SAE Journal*, Vol. 7, August, 1920, pp. 149-154: "Methods of Generating Involute Spline and Its Application," by D. T. Hamilton.



■ Fig. 2 – *K* for standard and full fillet type of splines

The SAE Standards Committee in January, 1920, decided to increase the number of splines in shafts, which brought out the following comment from the same paper:

"But in all the present SAE standards the same number of splines is used for the various diameters, which seems to be not exactly what the requirements demand. It would apparently be much better to have the number of splines governed by the diameter. In other words, the number of splines should be greater as the diameter increases. This

proposal has an advantage in the fact that the greater the diameter, as a general rule, the greater the torque the part is required to carry; consequently, a greater number of driving points is necessary. Two additional advantages are presented by the consequent reduction in wall thickness and the possibility of facilitating interchangeable manufacture."

The author further presented proposals for 10-spline shafts up to 1.75 diameter, 12 to 2.50, 14 to 3.50, 16 to 4.50, 18 to 5.50, and 20 for 6.00 diameter. The pitches were all odd, and the length of spline recommended was from 0.67 to 0.72 of the diameter.

Involute splines, because of the greater accuracy of machining, may be fitted with closer limits. The author says further:

"These straight splines are produced by regular machining methods, and are then ground after hardening in the same cases, which gives fairly accurate results. However, the only grinding that can be done on the gears is in the hole; therefore it is necessary to provide considerable tolerance on the splines so that they can slide freely. In order that the members shall slide freely, the gear cannot fit too closely on the shaft and, where there is considerable tolerance between the splined members, they are subject to vibration when under load, which results in noisy gearing. The spline shaft is superior to the square shaft in this connection. Splines having involute sides provide more bearing surface and permit closer fits, thus presenting a further improvement."

In summing up the advantages of the involute spline, the last paragraph in Mr. Hamilton's paper is quoted:

"The foregoing contains only a brief outline of the advantages of involute splines. Briefly summarized, they can be said to possess greater possibilities for securing interchangeability between parts; the involute can be generated much more accurately, because the tool used in producing it can be generated after hardening; the involute has a tendency to keep the parts concentric with each other, and when used in an automobile transmission this would have the advantage of decreasing the noise of the gears. Further, the number of standards necessary to cover all conditions is considerably reduced, resulting in lower tool and manufacturing costs."

The transmission of power from a shaft to a gear, or vice versa, has caused the development of several types of keys, depending on the stress involved and the working conditions. Some gears must slide while transmitting power, some must be susceptible to shifting under no load, and some are pressed on permanently. Loose keys cannot be used except for permanent fits, and even they cannot be used when high stresses are apt to develop fatigue weakness at a sharp corner or minimum section. This weakness has forced the development of multiple keyways, or splines, in which each keyway is made much more shallow than in the single-key design. This results in an increase in diameter of the minimum section, and, at the same time, presents more surface contact between driving and driven members. Millions of automobile transmissions have splined shafts where gears are shifted under no load.

Plain splines with parallel sides have been used under many conditions, and have been very satisfactory in spite of some difficulties in manufacturing. The automobile transmission started out to use the square shaft, then adopted the four-spline, and later the common practice was to use the six- or ten-spline shaft. High stresses in shafts, especially in aircraft engines, have resulted in failure at the root of the splines, aggravated by the sharp corner left in the machining. Attempts to incorporate a fillet at the root resulted in loss of tooth contact, and the next step was to use an involute form of spline which incorporates a fillet at the root, formed by the generating action of the cutter or hob. This fillet can be made large or small. The smallest fillet will result from using a hob or shaper cutter with a sharp corner, and the largest fillet with a hob or cutter having a large radius or chamfer at the corner. This inherent feature in the involute spline has led to its adoption in the aircraft-engine industry nearly to the exclusion of all other types. The fact that the involute spline is self-centering and can be accurately machined without grinding adds to the reasons for its popularity. Since a single hob will cut any number of teeth of a given pitch on the shaft, this is an economy not to be overlooked, but the important feature is the ability to cut splined shafts without delay from available hobs.

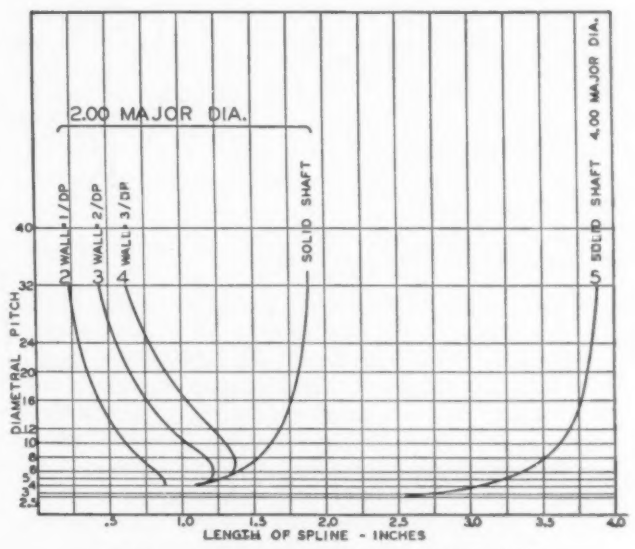
For the internal, available shaper cutters can be used over a range of teeth, so there is some flexibility for experimental work. With some exceptions, broaches are used in mass production for the internal spline, especially for small diameters.

Since multiple splines divide the load among a number of teeth, hopefully the full number, they need not be as long as a single key might be. If the load is to be divided equally, each spline may be designed to carry its share of the load, with two considerations: first, the fiber stress at the root; and second, the tooth contact area. Either one may be the controlling factor, depending on operating conditions, but the torque capacity of neither could exceed the capacity of the shaft on which the spline teeth are cut.

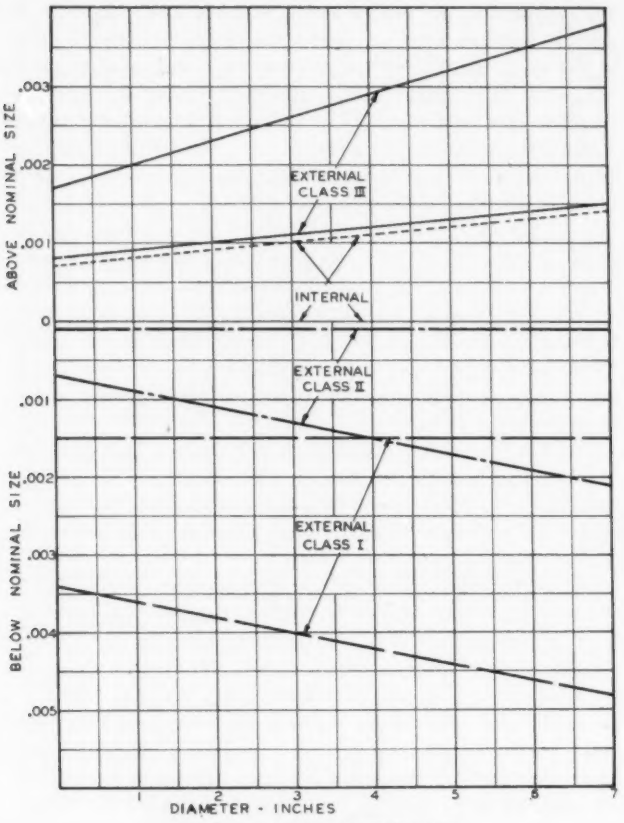
■ Comparison of Straight and Involute Splines

A comparison of straight and involute splines will bring out some interesting data. A very popular size in the automobile industry is the 1.250 OD – 6 tooth, to slide when not under load (SAE 6B). With this we will list an 8/16 DP with 9 teeth:

	SAE-6B	8/16 DP Involute
Major Diameter	1.250	1.250
Minor Diameter	1.062	1.000
Tooth Area Per In. Length	0.5625	0.890
Fillet on Minor Diameter	None	0.016



■ Fig. 3 – Spline length – solid and hollow shafts



■ Fig. 4 – Major diameter fits

In the above comparison, the minor diameter of the involute is slightly less, but the tooth contact is more than in the straight spline. Another comparison, of the 2-in. diameter SAE-6B and an 8/16 DP-15-tooth spline, shows a reversal, that is, the straight having a minor diameter of 1.700 against the involute's 1.7500. This illustrates the flexibility of the involute design, as there is always a choice of diametral pitch and diameter to meet an individual condition.

Torsional tests of plain splined shafts have indicated that their strength is equivalent to a plain shaft of slightly less than the minor diameter. Failures were attributed to the sharp corners at the root of the spline. In the involute type of splined shaft, tests have shown it to be the equivalent of a shaft larger than the minor diameter, indicating that with the fillet, the spline teeth actually add strength under fatigue loading.

■ Flexibility of Design

Sometimes the designer has complete control of the diameter of shaft to be used, but often has to accommodate a ball bearing or other fixed part supplied in predetermined sizes. With the old spline standard, there were sizes every 5 mm step in diameter, and with the straight spline there were steps of ⅛ in. from 0.750 diameter up.

In the new standard proposed for involute splines, almost any diameter can be found in some diametral pitch. Fig. 1 shows a line for diameters from 0.375 to 4.000 and the choice of numbers of teeth and pitches throughout the range. For any given DP, the number of teeth between one diameter and the next largest shown represents the number of choices of diameter, and if there are not enough in one pitch, the next finer pitch will have even more choices. The whole is a graphic representation of the tables in the standard.

■ General

The letters and symbols used in this paper are the same as in the proposed standard for involute splines, and any references to the standard will be understood to be the proposed standard for involute splines, copies of which have been sent out to American industry.

The symbols used were taken from the ASA standard wherever applicable. Since, with splines there is no center distance, C was used for measurements over pins, and G, being unassigned, was used for measurements between pins. These symbols are:

DP = Diametral Pitch
p = Circular pitch
a = Addendum
b = Dedendum
h_k = Working depth
c = Clearance
t = Circular tooth thickness
t_s = Circular space width
ϕ = Pressure angle
C = Measurement over pins – external
G = Measurement between pins – internal
N = Number of teeth
D = Pitch diameter
D_b = Base circle diameter
D_o = Major diameter
D_R = Minor diameter
TIF = Diameter at junction of involute form with fillet
d_s = Measuring pin diameter – external
d_a = Measuring pin diameter – internal

■ Involute Spline Capacity

The torque capacity of a splined shaft has been discussed many times in relation to a simple torsional force and as installed where fatigue loads will induce failure. Under either condition, the shaft with a sharp corner at the base of the spline will fail more quickly than the one with a carefully formed fillet. Straight splines not only have sharp corners, but often have a grinding relief which cuts under the normal minor diameter. The involute spline has normally a small fillet at the base of the tooth, which can easily be made larger by chamfering or rounding the tip of the hob. A shaft so made has been found to have slightly more strength, torsionally, than a plain shaft whose diameter is the same as the spline shaft minor diameter.

Conservatively, therefore, it seems that, from a strength consideration, the torque capacity of a splined shaft would be:

$$Q = \frac{\pi D_R^3 S_s}{16} \qquad (1.6)$$

Fig. 2 shows the factor $\frac{\pi D_R^3}{16}$ for the standard and full fillet type of splines. It shows that the torque capacity of a full fillet type of spline is less than the standard flat base, due to the smaller diameter at the root of the tooth for the same major diameter, but this, in practice, would be offset by the use of a high-strength alloy steel for applications involving fatigue loads.

The capacity of a splined shaft is limited either by the pressure of one tooth or a series of teeth against another series of teeth, or by the shear load on the tooth itself.

If A_t is the contact area per tooth, it is the product of the height h and length L. Since in the involute standard

$$h = \frac{0.8}{DP} = \frac{0.8D}{N}$$

(0.80 is a minimum) (footnote 2):

$$A_t = \frac{0.8LD}{N}$$

The amount of torque a splined shaft can carry, assuming all teeth contacting is:

$$Q = \frac{S_c A_t N D}{2} = 0.4 S_c L D^2 \qquad (1.7)$$

where S_c is the allowable compressive bearing stress.

For shear stress at the pitch diameter, the area in shear will be $\frac{\pi DL}{2}$, and the torque transmitted will be:

$$Q = \frac{\pi DL}{2} \times S_s \times \frac{D}{2} = \frac{\pi L D^2 S_s}{4} \qquad (1.9)$$

This averages for the internal and external splines.

Since both these formulas represent the torque a shaft may safely transmit, we may place them equal, thus:

$$\frac{\pi L D^2 S_s}{4} = 0.4\, S_c L D^2$$

and simplifying obtain:

$$\frac{S_s}{S_c} = \frac{1.6}{\pi} = 0.509$$

ratio of the two stresses.

Good practice[3] shows that the shear stress allowable is 60% and the compressive bearing stress is 140% of the tensile ultimate strength, which means a ratio of S_s/S_c of 60,000/140,000 or 0.428. Since this is so, the ratio will never reach 0.509, and the shear load S_s must be considered the critical one and all calculations should be checked by formulas involving only S_s. If we now combine formulas (1.9) and (1.6) we will determine a desirable length as the shear load cancels out.

Thus:

$$\frac{\pi L D^2 S_s}{4} = \frac{\pi D_R^3 S_s}{16}$$

or

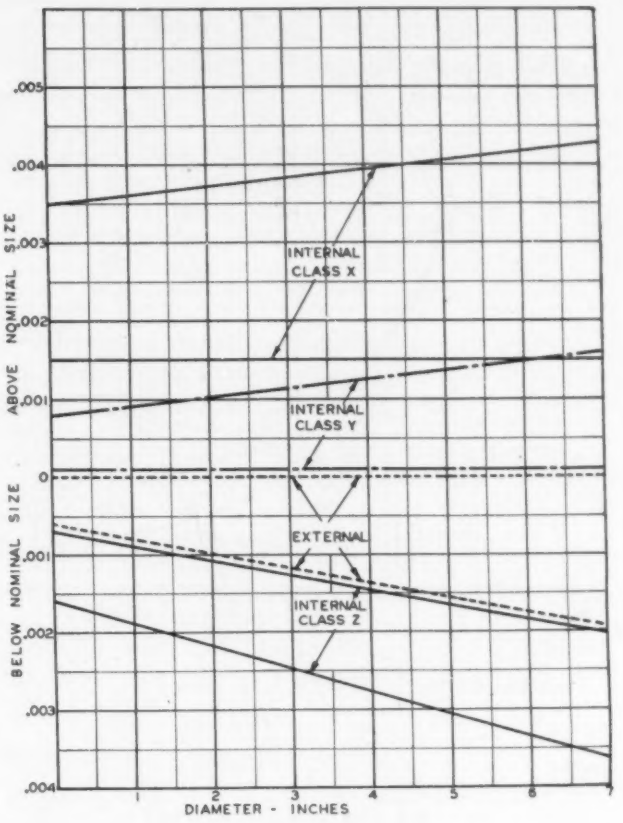
$$L = \frac{D_R^3}{4D^2} = 0.250 \frac{D_R^3}{D^2} \qquad (2.0)$$

This tells us that if all teeth are in perfect contact, a spline of length L would transmit all the torque a shaft of D_R diameter will carry. Since perfect parts cannot be made, even when lapped, it is good practice to assume that errors of spacing and form will allow 25% of teeth to contact, and this would mean multiplying formula (2.0) by 4 and obtaining:

$$L = \frac{D_R^3}{D^2} \qquad (2.1)$$

[2] In the standard, h varies from 0.779 for $N = 6$ to 0.833 for $N = 50$, being 0.802 for $N = 9$. This will cause a variation of ratio S_s/S_c from 0.496 to 0.529.

[3] Specification ANC-5.



■ Fig. 5 – Minor diameter fits

which is a good approximate formula for any spline on a solid shaft.

All splines should, therefore, be made of length L, in accordance with formula (2.1) for solid shafts.

In order to illustrate the effect of changing DP on formula (2.1) Fig. 3 is presented. Each example taken has the same major diameter in each series. This shows that as the pitch becomes finer the length L approaches the pitch diameter and is always less than the major diameter. This refers to solid shafts, and for hollow shafts where the torque capacity is less, the tooth length required would be less.

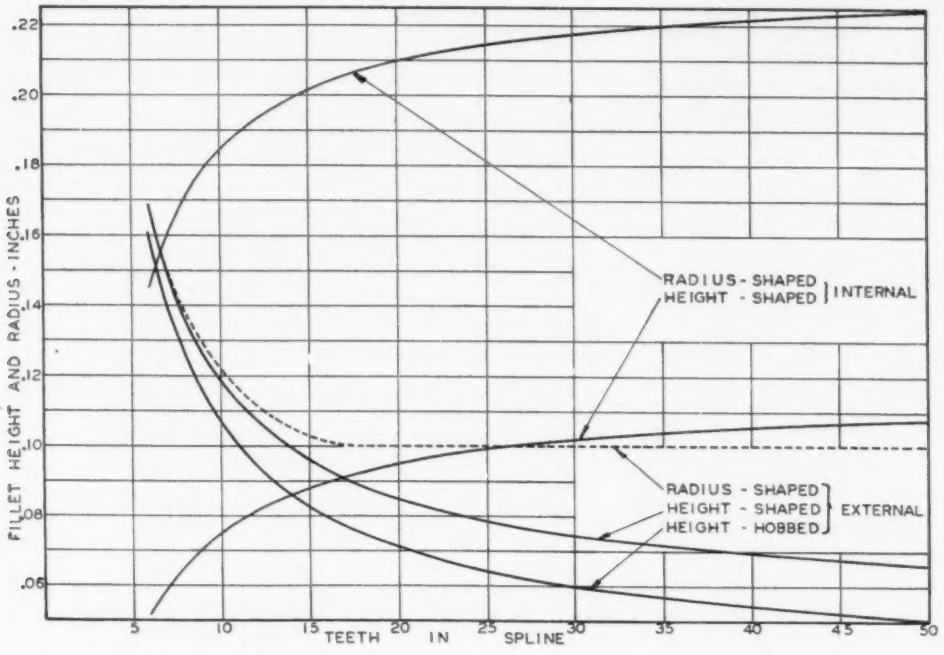
For a hollow shaft, the torque capacity at D_R will be:

$$Q = \frac{\pi (D_R^4 - D_1^4) S_s}{16 D_R}$$

and if we combine this with (1.9), we can obtain

$$L = \frac{(D_R^4 - D_1^4)}{4 D^2 D_R} \qquad (2.2)$$

Splined shafts of large diameter are usually made with a rim design like a gear, or the splines are cut on a sleeve or hollow shaft. The rim should be of sufficient thickness to resist distortion during machining, and stronger than the teeth. An exact analysis of the stresses involved is too complex to introduce here and involves many assumptions of an arbitrary nature. Good proportioning will result from making the rim thickness at least equal to the thickness of the tooth at its base.



■ Fig. 6—Tooth fillets

Formulas for tooth thickness t_2, at base, are:

$$\text{Cos } \phi = \frac{D_b}{D_R}$$

and

$$t_2 = D\left(\frac{t}{D} + \text{INV } \phi - \text{INV } \phi_4\right)$$

and by using these we can develop Table 1.

Table 1 – Tooth Base Width, ½ DP

N	Cos $\phi_4 = \frac{D_b}{D_R}$	INV ϕ_4	
10	0.9623	0.0072	1.8327
20	0.9116	0.0273	1.9948
30	0.8959	0.0355	2.0466
40	0.8882	0.0399	2.0722
50	0.8837	0.0426	2.0875

If the rim thickness is made so that it is stronger than the tooth at its base, as shown in Table 1, it should have sufficient rigidity and strength to carry the maximum shear load which could fall upon any one tooth. On this basis, it is recommended that the rim thickness be from 2.00 to 2.25 times the tooth depth, or otherwise stated as $\frac{2.00}{DP}$ to $\frac{2.25}{DP}$. If the rim thickness is made $\frac{1}{DP}$, the (2.2) formula would then become:

$$L = \frac{D_R^4 - \left(D_R - \frac{2}{DP}\right)^4}{4D^2 D_R}$$

To compensate for inaccuracies, we must, as above, multiply by 4, which makes the formula:

$$L = \frac{D_R^4 - \left(D_R - \frac{2}{DP}\right)^4}{D^2 D_R} \qquad (2.3)$$

Assuming again a shaft having a major diameter of 2.00, of varying diametral pitches, we can develop a curve, No. 2 in Fig. 3, which shows the spline length for a wall thickness of 1/DP. Curve No. 3 is the same kind of a curve with the wall thickness equal to 2/DP, and curve No. 4 shows the spline length when used with wall thickness of 3/DP. For a comparison, curve No. 5 shows the spline length needed on a 4.00 major diameter solid shaft.

Let us assume that a shaft must carry a torque of 400 ft-lb, and we wish to determine the choice available and whether to pick a coarse or fine pitch. By formula (1.6):

$$12 \times 400 = \frac{\pi D_R^3 S_s}{16}$$

and if we assume S_s to have a maximum value of 60,000, then $D_R = 0.74$.

By consulting the tables, we find that we can use the following, in each of which $D_R = 0.75$:

DP	N	D	D_s
8/16	7	0.875	0.983
12/24	10	0.8333	0.904
16/32	13	0.8125	0.864
20/40	16	0.8000	0.841
24/48	19	0.7917	0.824

Using formula (2.1), we find that:

$$L = \frac{D_R^3}{D^2} = \frac{0.423}{D^3}$$

showing that the greater D is, the less L need be; therefore, the coarser pitch should be selected.

By the above formula, therefore:

DP	L
8/16	0.553
12/24	0.608
16/32	0.642
20/40	0.661
24/48	0.675

■ Methods of Fitting Splined Parts

There are three ways of fitting splined parts:

1. On the major diameter.

2. On the sides of the teeth.
3. On the minor diameter.

Major diameter fits are most generally used, as the major diameter is usually accurately held, probably by grinding, and often is used as a pilot or locating point for other dimensions. The broach which cuts the internal teeth is accurately made on the major diameter, and we, therefore, have good reasons for using the major diameter to control fit.

Since the involute form must be accurately cut on both internal and external, it is, therefore, a good fitting point. Errors of spacing sometimes make such a fit difficult, and these errors, combined with errors of cutting the involute form, sometimes introduce an eccentricity not present in the major diameter fit. For the full fillet form of tooth, however, this fit is the only one which can be used, as there is a large clearance at both major and minor diameter.

The minor diameter fit is used where the manufacturer favors grinding the minor diameter of the internal spline to vary the fit on a shaft which is held standard. It is particularly adapted to the straight spline, where the minor diameter of the external (shaft) can be form ground to contact the ground minor diameter of the internal spline.

Either major or minor diameter fits are comparable with plain cylindrical fits and may follow well-established practice, but side bearing fits have other complications. If tooth spacing and form were perfect, these fits could be treated as a cylindrical fit, but errors of cutting will sometimes change a free fit to a tight fit, so it is good practice to allow more clearance for this type.

Major and Minor Diameter Fits

When the subcommittees took up the question of fits on the major and minor diameters, there were two groups vitally interested, one for the involute, and the other for the straight spline standardization. In January of 1945 they met together and agreed on three classes of fits for 1.25 diameter:

Sliding	0.0015 loose to 0.0045 loose
Close	0.0001 loose to 0.0018 loose
Press	0.0001 tight to 0.0021 tight

Further, all fits are to be varied with diameter on the basis of 0.0003 per in. of diameter, with the extra allowance on the free side for clearance, and on the tight side for interference or press fits. On this basis charts were made, and are presented here as a record.

Fig. 4 shows the major diameter fits, based on size of internal which varies 0.0001 per in. of diameter on the tolerance. Each of the external splines has a tolerance with a variation of 0.0002 per in. of diameter, to make a total of 0.0003.

Fig. 5 shows the minor diameter fits, just the same as that for the major diameter chart, except that the external, as the standard or unchangeable member, has a 0.0002 variation per in. of diameter, and the internal in each case has a variation of 0.0001, making a total of 0.0003 per in. of diameter.

Limiting assembly to one position may be accomplished by making the external spline in the usual way and afterward milling out one tooth. The broach is then made with one tooth omitted, which leaves a wide tooth in the product to fit the wide space in the external spline. An alternate method is to fill up a space in the internal spline, which can be done by pressing a pin through the rim into the space.

Developing the Tooth Form

The working part of the tooth of both external and internal splines is a perfect involute. The total depth of the tooth is equal to 1/DP, or the addendum is 0.5/DP. This stub tooth form has been called the one-two system, due to the fact that it was called ½ DP, or 10/20 DP, the secondary pitch in the denominator being twice that in the numerator.

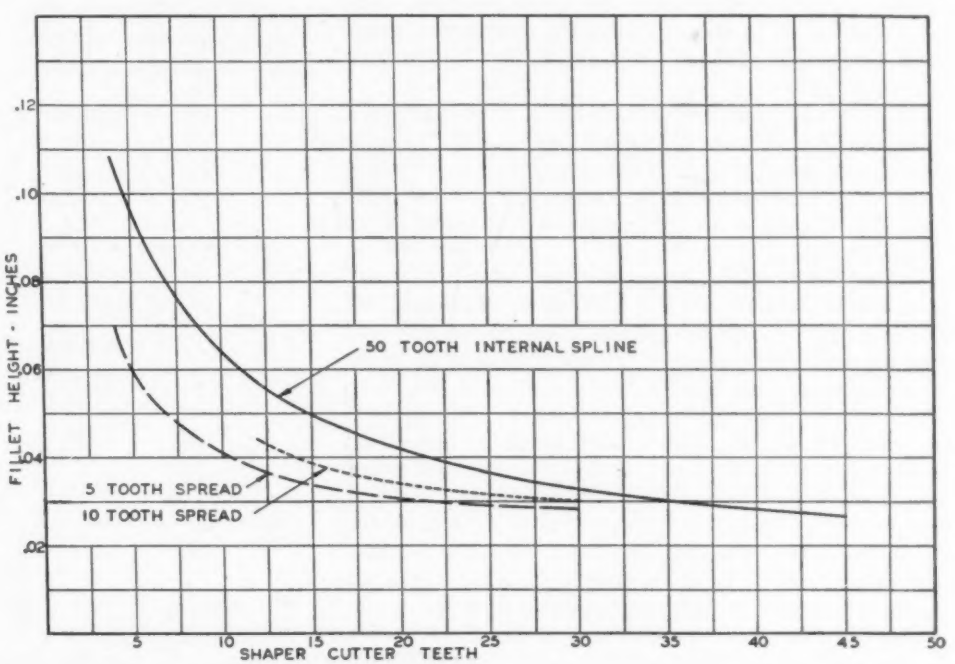
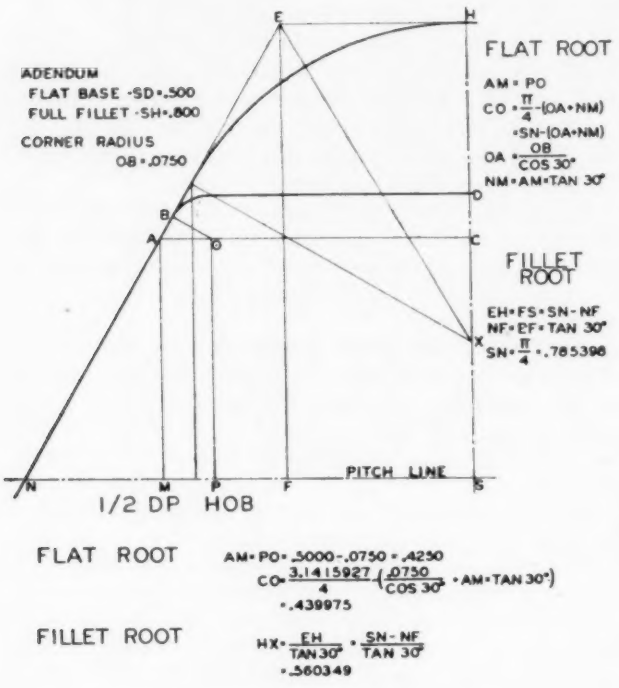


Fig. 7 – Fillets formed by shaper cutters



■ Fig. 8 – Outline of hob tooth for flat root and fillet root types

The master hob tooth is one having a tooth thickness equal to 0.50 of the basic tooth circular thickness, and addendum of 0.5/DP, a tip radius of 0.020944 times the tooth thickness, or 0.075/DP, and a ramp at the root to chamfer the tip of the product tooth. The shaper cutter has the same tooth thickness and addendum, but has a chamfer at the tip of 0.025/DP in height. These corners are considered a minimum in well-established gear-cutting practice, and in the standard, the fillets given are those which are cut with these tools. Hobs and shaper cutters for the full fillet type have a larger radius or chamfer to fit the requirements.

The point at which the involute curve joins the fillet is called the "true involute form diameter" or TIF. This, for the internal spline must be slightly larger than the major diameter of the external spline, and for the external spline it must be slightly smaller than the minor diameter of the internal spline.

During the development of the standard, S. O. Bjornberg, Illinois Tool Works, and C. R. Staub, Michigan Tool Co., carried out studies to determine the proper fillet and fillet height and the angle of chamfer of the spline teeth. Fig. 6 shows the fillet and the height of fillet for all splines. The fillet chosen for the standard was always the greatest one; in this case it was that required for the shaped spline. Since this height is the basis for the determination of the TIF, it was important to work to the maximum dimension expected with existing production methods. The curve for the shaped spline was used as a basis for external spline fillet height, and the radius was taken from the large-scale layouts which were made for each number of teeth.

The fillets for the major diameter of the internal were also worked out to a large scale, and the curves show the fillet developed by a shaper cutter of standard design. The broach could be made to cut a smaller fillet, but again, the standard was set to allow the maximum needs of existing equipment.

If an internal is cut with a shaper, the number of teeth in the cutter affects the fillet, because of the difference in generating action. Fig. 7 shows the fillet height generated in a 50-tooth internal spline by cutters of varying number of teeth. A curve on this graph (Illinois Tool Works) shows the fillet height generated by a cutter having five teeth less than the part being cut, and another shows that generated by a cutter having 10 teeth less than the part.

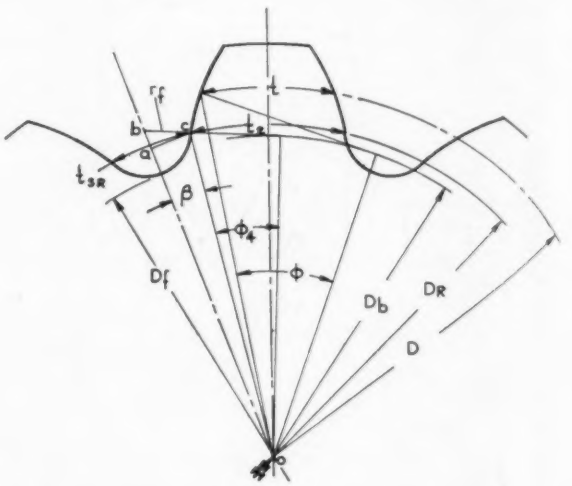
There probably is no best combination of cutter and work, but in the standard (Appendix 1) there is a table for guidance. Manufacturing conditions are never identical in all details in two different places, so consultation with the tool engineer is always a wise precaution.

Fig. 8 shows the outline for the hob tooth for flat root and fillet root types. The mathematical solution for dimensions is included, and shows the distance to the center of the 0.0750 radius for the flat root, and for the fillet root type the radius required to join the two flanks with the addendum of 0.8000. If it is desired to grind the hob so that the radius comes slightly outside tangency with the flank, the center distance *CO* should be increased slightly, and for the fillet type the radius *HX* should be slightly greater than that calculated. This effect, which is termed "capping," should be allowed the hob maker if he so wishes, to avoid the radius cutting into the flank of the hob.

■ Full Fillet Spline

Involute splines which are cut with a full fillet at the base have a minor diameter less than the normal or flat base type, but due to the smooth large fillet are less liable to fatigue failure, and, therefore, will safely carry the same torque as a plain shaft whose diameter is slightly larger. This has been proved by several engineers, and the full fillet type has, therefore, been used extensively in the aircraft-engine industry.

This type of spline, except for the broached internal, cannot be cut to an exact radius due to the generating action of a shaper cutter or that of a hob. The radius can only be given, therefore, as an approximate one, so far as



■ Fig. 9 – External spline

inspection of a part is concerned. It is safe to say that such a fillet will not average more than the radius which will be just tangent to the involute tooth form at the true involute form (TIF) diameter. The generating action may, however, cut below the root determined by this radius, when the hob is fed deeper to vary the fit on the spline teeth.

For an idealistic approach, we may calculate the fillet except for the 6- and 7-tooth external, in which the minor diameter is less than the base circle. These calculations are formed around basic conditions. See Fig. 9 for the external spline.

D_R = Minor diameter, basic
ϕ_4 = Pressure angle at D_R
t_2 = Tooth thickness at D_R
r_f = Fillet radius
D_f = Diameter at base of fillet

$$\text{Cos } \phi_4 = \frac{D \cos \phi}{D_R} = \frac{D_b}{D_R}$$

$$t_2 = D_R \left(\frac{t}{D} + \text{INV } \phi - \text{INV } \phi_4 \right)$$

$$t_{sR} = \frac{\pi D_R}{N} - t_2 \qquad \text{Arc } \beta = \frac{t_{sR}}{D_R} \text{ (radians)}$$

$$ac = \frac{D_R \sin \beta}{2}$$

$$bc = r_f = \frac{ac}{\cos (\phi_4 + \beta)} = \frac{t_{sR}}{2 \cos (\phi_4 + \beta)} \text{ (approximately)}$$

$$D_f = 2 (oa + ab - bc)$$

$$= 2 \left[\frac{D_R \cos \beta}{2} + \frac{t_{sR} \tan (\phi_4 + \beta)}{2} - \frac{t_{sR}}{2 \cos (\phi_4 + \beta)} \right]$$

$$= D_R \cos \beta + t_{sR} \tan (\phi_4 + \beta) - \frac{t_{sR}}{\cos (\phi_4 + \beta)} \text{ (approximately)}$$

See Fig. 10 for the internal spline.

ϕ_5 = Pressure angle at D_o
t_3 = Circular tooth thickness at D_o
D_{mR} = Major diameter at root

$$\text{Cos } \phi_5 = \frac{D \cos \phi}{D_o} = \frac{D_b}{D_o}$$

$$t_3 = D_o \left(\frac{t}{D} + \text{INV } \phi_5 - \text{INV } \phi \right)$$

$$t_{so} = \frac{\pi D_o}{N} - t_3 \qquad \text{Arc } \beta_1 = \frac{t_{so}}{D_o} \text{ (radians)}$$

$$ac = \frac{D_o \sin \beta_1}{2} = \text{Half chordal width of space } t_{so}$$

$$r_f = bc = \frac{ac}{\cos (\phi_5 - \beta_1)} = \frac{D_o \sin \beta_1}{2 \cos (\phi_5 - \beta_1)}$$

$$D_{mR} = 2 (ob + bc) = 2 \left[\frac{D_b}{2 \cos (\phi_5 - \beta_1)} + \frac{ac}{\cos (\phi_5 - \beta_1)} \right]$$

$$= \frac{D_b + t_{so}}{\cos (\phi_5 - \beta_1)}$$

where $t_{so} = 2ac$ approximately.

By means of the above formulas, the fillet for any tooth thickness may be calculated, but usually the approximate fillet is desired.

The fillet actually formed by the generating action of a cutter or hob will be an irregular curve. Considerable variation on the root diameter may be allowed within the tolerances dictated by stress, as long as the fillet is free of tool marks and relatively smooth to avoid stress raisers at any point between the TIF diameter on one tooth and the corresponding point opposite on the next tooth.

The above is an idealistic approach, but in actual practice a single hob is used for all tooth numbers of a given pitch, and a standard tooth design used for all shaper cutters, such that the dedendum is the same for all splines of a given DP, as in the flat base involute spline.

A hob made with a perfect radius (0.560) on the tip, to cut a dedendum of 0.8/DP will, in the generating action, develop a radius a small percentage greater than the hob radius. Likewise a cutter, in an internal spline, will develop a slightly larger radius on the part than that on the cutter (see Fig. 11).

In the tables, the major diameter of internal splines and minor diameter of the external splines are given as basic; that is, pitch diameter ± 1.600/DP. If it is desired to determine exactly what the minor diameter of the external spline would be, use the following formula:

$$D_f = \text{PD} - \frac{1.600}{\text{DP}} - \text{circular tooth clearance} \times E \text{ (Table IV)}$$

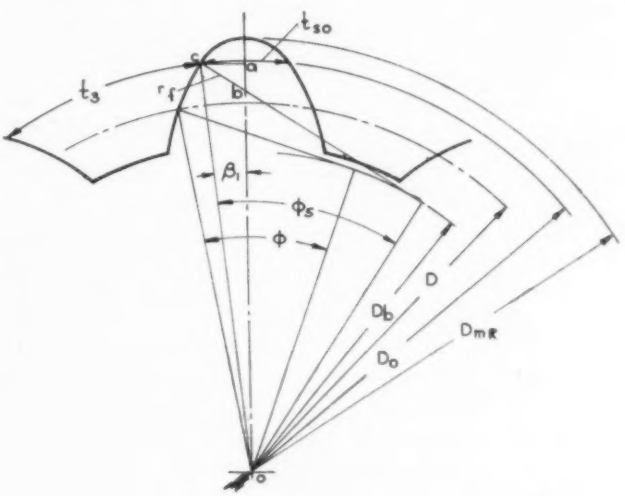
or

D_f = Basic diameter (Column 17 Table XX or XXI) − circular tooth clearance × E

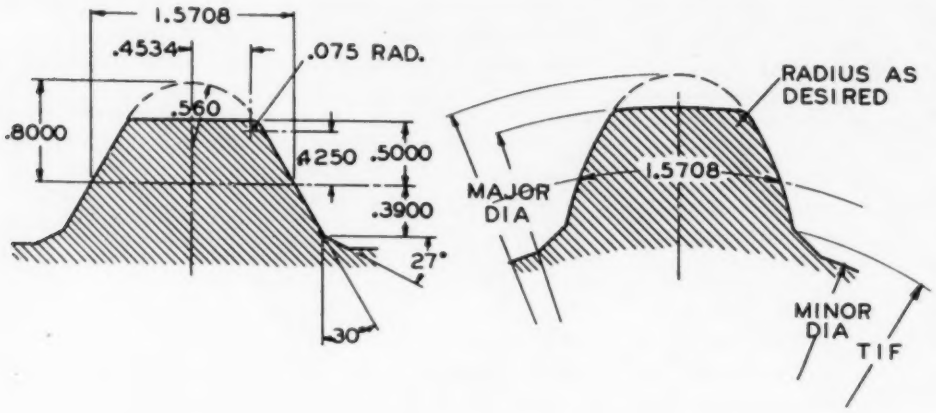
This takes into consideration the fact that the shaft (external spline) is always varied to control the fit by forcing the hob to cut to that depth which will obtain the desired side bearing fit.

■ Measuring Splined Fittings

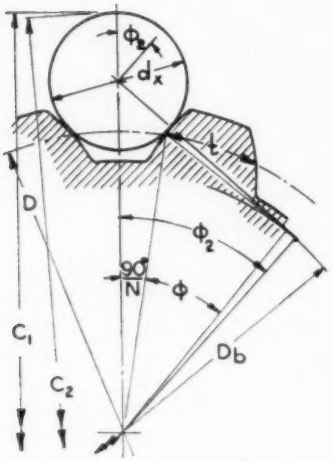
The most accurate means of checking a splined part for size is that used for involute gears; that is, with pins. The two pins used contact the involute tooth form, and the micrometer measurement, between the pins for internal splines and over pins for the external splines, is a check of the space or tooth width. This method is too slow, however, for mass production, and does not check errors in tooth spacing or involute form; therefore, plug and ring gages are used which do detect spacing errors, and are much faster to use. These gages must be checked with pins and space checkers, so that the pin method is basically



■ Fig. 10 – Internal spline



■ Fig. 11 – Left: ½ DP hob tooth [Fig. 6]; right: ½ DP broach tooth [Fig. 7]



Φ = 30° = PRESSURE ANGLE AT D
Φ_2 = PRESSURE ANGLE AT PIN CENTER
d_x = PIN DIAMETER
D_b = D COS 30°
t = CIRCULAR TOOTH THICKNESS

$$\text{INV}\,\Phi_2 = \frac{t}{D} + \text{INV}\,30° + \frac{d_x}{D_b} - \frac{\pi}{N} \qquad (1)$$

FROM INV Φ_2 FIND Φ_2 IN DEGREES

IF "N" IS EVEN

$$C_1 = \frac{D_b}{\cos \Phi_2} + d_x \qquad (1.1)$$

IF "N" IS ODD

$$C_2 = \frac{D_b \cos \frac{90°}{N}}{\cos \Phi_2} + d_x \qquad (1.2)$$

EXAMPLE - FOR 10/20 P, N=20, $t = t_s - .001$

$$\text{INV}\,\Phi_2 = \frac{.1561}{2.000} + .053751 + .110851 - .157080$$

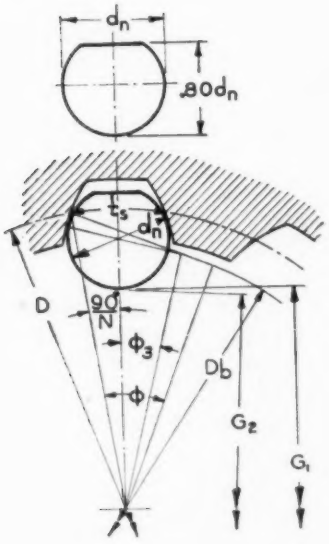
$\text{INV}\,\Phi_2 = .085573$ $\quad \Phi_2 = 34.552°$

(DIMENSIONS FROM TABLE IV)

USE (1.1) BECAUSE "N" IS EVEN

$$C_1 = \frac{1.732051}{\cos 34.552°} + .1920 = 2.2950$$

■ Fig. 12 – External spline and formulas [Fig. 4]



Φ_3 = PRESSURE ANGLE AT PIN CENTER
t_s = CIRCULAR TOOTH SPACE (BASIC)

DUE TO INTERFERENCE WITH MAJOR DIAMETER, PIN MUST BE FLATTENED AS SHOWN.

$$\text{INV}\,\Phi_3 = \text{INV}\,30° + \frac{t_s}{D} - \frac{d_n}{D_b} \qquad (1.3)$$

IF "N" IS EVEN

$$G_1 = \frac{D_b}{\cos \Phi_2} - d_n \qquad (1.4)$$

IF "N" IS ODD

$$G_2 = \frac{D_b \cos \frac{90°}{N}}{\cos \Phi_3} - d_n \qquad (1.5)$$

EXAMPLE - FOR 10/20 P, N = 20

$$\text{INV}\,\Phi_3 = .053751 + .078540 - .083138$$

$\text{INV}\,\Phi_3 = .049153$ $\quad \Phi_3 = 29.183°$

(DIMENSIONS FROM TABLE IV)

USE (1.4) BECAUSE "N" IS EVEN

$$G_1 = \frac{1.732051}{\cos 29.183°} - .1440 = 1.8399.$$

■ Fig. 13 – Internal spline and formulas [Fig. 5]

the most direct way of defining measurements of space and tooth thickness for mating parts. For the benefit of the gage maker, it is advisable, in the specifications, to indicate how much variation may be allowed in both spacing and involute form.

Earle Buckingham, M.I.T., has developed formulas for gear measurements, which can be adapted to the measurements of spline fittings. His formula No. 11, with the symbols as adapted to spline measurements, for external is:

$$\text{INV } \phi_2 = \text{INV } \phi + \frac{t}{D} + \frac{d_s}{D \cos \phi} - \frac{\pi}{N} \qquad (1)$$

and his No. 12 and No. 14 for measurements over pins, as adapted to the spline symbols are:

$$C_1 = \frac{D_b}{\cos \phi_2} + d_s \qquad (1.1)$$

for even number of teeth.

$$C_2 = \frac{D_b \cos \frac{90}{N}}{\cos \phi_2} + d_s \qquad (1.2)$$

for odd number of teeth.

See Fig. 12.

For the internal spline, Buckingham's formula No. 19, when modified to measure space instead of tooth thickness, becomes:

$$\text{INV } \phi_3 = \text{INV } \phi + \frac{t_s}{D} - \frac{d_n}{D_b} \qquad (1.3)$$

and his formulas No. 20 and No. 21 become:

$$G_1 = \frac{D_b}{\cos \phi_3} - d_n \qquad (1.4)$$

for even numbers of teeth.

$$G_2 = \frac{D_b \cos \frac{90}{N}}{\cos \phi_3} - d \qquad (1.5)$$

or odd numbers of teeth.

See Fig. 13.

The theoretical pin sizes for basic conditions to contact at the pitch line, on both internal and external splines, can be readily calculated by modifying formulas (1) and (1.3). If the contact is at the pitch line, then ϕ_2 for the external will be equal to $\phi + \frac{90}{N}$, and for the internal spline ϕ_3 will be equal to $\phi - \frac{90}{N}$.

Substituting these values in formulas (1) and (1.3), we obtain:

$$d_s = D_b \left[\text{INV} \left(\phi + \frac{90}{N} \right) - \text{INV } \phi + \frac{t}{D} \right]$$

$$d_n = D_b \left[\text{INV } \phi - \text{INV} \left(\phi - \frac{90}{N} \right) + \frac{t}{D} \right]$$

Using these formulas, we can find that:

$N = 6$	$d_s = 2.196$	$d_n = 1.607$
$N = 50$	$d_s = 1.848$	$d_n = 1.782$

These are theoretical sizes and are not obtainable.

Measuring pins must be accurate to the extreme, true in shape, and of the same size. They may contact the tooth side at the pitch circle or near it, but should not contact near the point of the tooth nor too far below the pitch circle. On both external and internal splines, an ideal pin will contact at the pitch circle at the design pressure angle, at a smaller angle below, and at a greater angle above. The best zone of contact is, therefore, from the pitch circle outward, but this cannot always be accomplished, because it would involve a different diameter for each tooth combination. Compromises are, therefore, used, and pins are made in inverse ratio to the diametral pitch and based on a selection of measuring pins for one/two pitch. Several systems have been developed completely by the VanKeuren Co. for different pressure angles, the most popular sizes being the 1.4400 and 1.7280. Recent additions are the 1.9200 and the 1.6800, and with these four series of sizes almost any gear or spline may be measured.

The ideal pin sizes are still not generally agreed upon. C. H. Havill, Bendix Aviation Corp., has calculated that the perfect size for the 6-tooth internal splines is 1.63965 and for the 12-tooth 1.91895, and recommends that for internal splines a 1.728 pin be used from 11 teeth up and a 1.550 pin from 6 to 10 teeth. He has also found that with a 1.680 pin diameter, the pin center is on the base circle at 6.867 teeth, which means that this size pin cannot be used for 6 teeth. A compromise would be to use a 1.44 pin for 5 and 6 teeth, 1.56 for 7 to 15 teeth, and 1.68 from 16 teeth up. This can and will be done by experienced gear manufacturers, but the standard has been compiled with the 1.440 pin for all internals, with data in the appendix for the use of the 1.680 pin. If this compromise does not meet all conditions, the basic data in Table IV and the formulas in Figs. 12 and 13 will enable anyone to figure measurements for any pin sizes. Fig. 14 shows different pin sizes in a 6-tooth, ½ DP internal spline.

■ Short Cuts

The general formula for the internal spline is:

$$\text{INV } \phi_3 = \text{INV } \phi + \frac{t_s}{D} - \frac{d_n}{D \cos \phi} \qquad (1.3)$$

where

$t_s = 0.5$ circular pitch

$d_n = 1.440$

$D = N$

$\text{Cos } \phi = 0.86602540$

and may be simplified to develop table dimensions for ½ DP:

$$(\text{For } d_n = 1.44)\ \text{INV } \phi_3 = 0.05375149 - \frac{0.09197246}{N}$$

$$(\text{For } d_n = 1.68)\ \text{INV } \phi_3 = 0.05375149 - \frac{0.36910059}{N}$$

For the external spline, the formula:

$$\text{INV } \phi_2 = \frac{t}{D} + \text{INV } \phi + \frac{d_s}{D \cos \phi} - \frac{\pi}{N} \qquad (1)$$

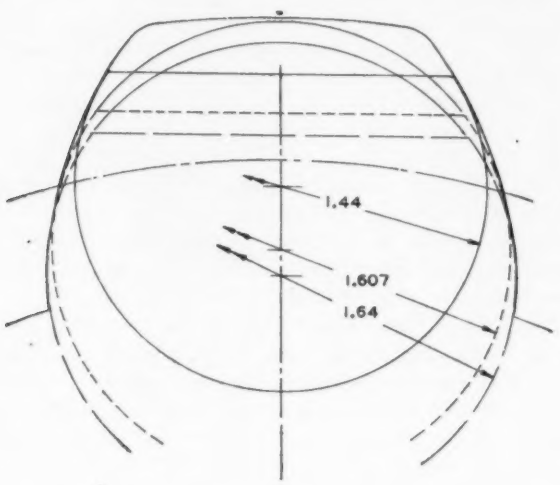
with basic dimensions as above, will become:

$$(\text{For } d_s = 1.92)\ \text{INV } \phi_2 = 0.05375149 + \frac{0.64622872}{N}$$

$$(\text{For } d_s = 1.728)\ \text{INV } \phi_2 = 0.05375149 + \frac{0.42452621}{N}$$

With the above simplified formulas, ϕ_2 or ϕ_3 can be quickly determined for ½ DP and will be inversely proportional for any other DP for basic conditions only, that is, with the tooth thickness one-half of the circular pitch.

The angles ϕ_2 and ϕ_3 vary between the values shown for 6 and 50 teeth, as indicated in Table 2.



■ Fig. 14 – Different pin sizes in 6-tooth, 1/2 DP internal spline

Table 2 – Variation in Values of ϕ_2 and ϕ_3

	Pin Diameter			
Internal:	1.44	1.68	1.728	1.92
ϕ_3 for 6 teeth, deg	27.03	8.30(N=7)	7.27(N=8)	....
ϕ_3 for 50 teeth, deg	29.68	28.66	28.44	....
External:				
ϕ_3 for 6 teeth, deg	....	....	38.59	41.57
ϕ_3 for 50 teeth, deg	....	....	31.38	32.04

■ Tolerances

For maximum and minimum tooth thickness, individual calculations may be made by formulas (1.0), (1.1), (1.2), (1.3), (1.4), and (1.5). Another and quicker method is to make the basic calculation, or use it from the standard tables, and add or subtract the tolerance times a factor E or F. These factors are obtained as follows:

External Spline Factor E – The original position of pin is shown in Fig. 15 (solid line and the secondary position taken when t has been increased is shown in dotted lines. Involutes are drawn through the pin center and are parallel, and x on the base circle is equal to the side opposite ϕ_2 in the small triangle at the pin center. Therefore:

$$x = \frac{\Delta C_1}{2} \sin \phi_2 \text{ (approximately)}$$

x at the pitch circle forms an angle with the tangent where Δt measurement applies. Therefore:

$$\frac{\Delta t}{2} = \frac{x}{\cos \phi} = \frac{\Delta C_1 \sin \phi_2}{2 \cos \phi}$$

and

$$\frac{\Delta C_1}{\Delta t} = \frac{\cos \phi}{\sin \phi_2} = E$$

and $\Delta C_1 = \Delta t \times E$

Example: DP = 8/16, N = 20, t = 0.1923

$\Delta t = t_s - t$ = 0.1963 − 0.1923 = 0.0040

C_1 = 2.8706 (Table XXVII), E = 1.52 (Table IV, Column 11)

ΔC_1 = 1.52 × 0.0040 = 0.0061

$C_s = C_1 - \Delta C_1$ = 2.8706 − 0.0061 = 2.8645

Internal Spline Factor F – The original position of pin is shown in Fig. 16 (solid line) and the secondary position taken when t_s has been increased is shown in dotted lines. Involutes are drawn through the pin center and are parallel, and x on the base circle is equal to the side opposite ϕ in the small triangle at the pin center. Therefore:

$$\Delta G_1 = \Delta t_s \frac{\cos \phi}{\sin \phi_2} = \Delta t_s \times F \text{ (Table IV, Column 7)}$$

Example: DP = 8/16, N = 20, $t_s = t_s - 0.0020$

G_1 = 2.2998 (Table XI, Column 14)

G_s = 2.2998 − 0.0020 × 1.80 = 2.2962

■ Flattening of Pins

The internal spline of the flat base or shallow type, where the whole depth of tooth is 1/DP, introduces another difficulty in the use of pins as the major diameter contacts the round pin before it can contact the involute tooth form on the two adjacent teeth. The amount, which is interference at the center of the space, can be calculated by the formulas:

$$\text{INV } \phi_3 = \text{INV } \phi + \frac{t_{sm}}{D} - \frac{d_n}{D_b}$$

$$\text{Interference} = 0.5 \left[\left(\frac{D_b}{\cos \phi_3} + d_n \right) - D_s \text{ min} \right]$$

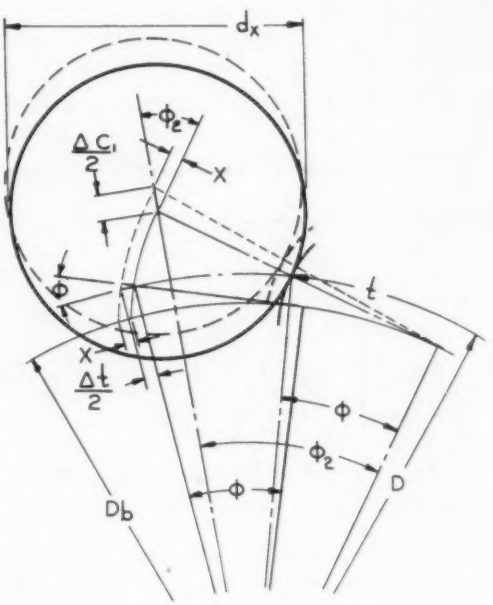
(If the result is negative, there will be a clearance.)

where

t_{sm} = Maximum space width

ϕ_3 = Pressure angle at pin center

To avoid the interference encountered with the use of the 1.44 pin, it must be flattened an amount sufficient to prevent the corner touching the involute or the fillet under extreme conditions. Referring to Fig. 17, if we assume the corner of the pin to be on the TIF diameter under extreme conditions, there will always be a clearance as the fillet starts at that point, and the corner is well outside the tangent point. In Fig. 17, the triangle formed by the three



■ Fig. 15 – Original position of pin shown as solid line; secondary position shown as dotted line – external spline [Fig. 10]

known sides of:

$$\left(B \text{ or } 0.5 \text{ TIF}\right)\left(\frac{G_1}{2}+\frac{d_n}{2}\right) \text{ and } \left(\frac{d_n}{2}\right)$$

can be solved to obtain x

$$\text{Cos } x = \frac{B^2 + (0.5\,G_1 + 0.5\,d_n)^2 - (0.5\,d_n)^2}{2B\,(0.5\,G_1 + 0.5\,d_n)}$$

For G_1 use table dimension [or calculate from formula (1.4)] plus maximum tolerance, hence for $N = 6$, DP = 1/2:

$$\text{Cos } x = \frac{(3.4485)^2 + (2.91855)^2 - (0.72)^2}{2 \times 3.4485 \times 2.91855} = 0.988198612$$

$$A = B \sin x = 3.4485 \times 0.15317801 = 0.52823436$$

$$C = \sqrt{0.72^2 - 0.52823^2} = 0.48925$$

$$\frac{C + 0.5\,d_n}{d_n} = \frac{1.2092}{1.44} = 0.839 \text{ flattening}$$

If we follow through the same calculation for $N = 50$

$$\text{Cos } x = \frac{(25.3955)^2 + (24.9217)^2 - (0.72)^2}{2 \times 25.3955 \times 24.9217} = 0.999767803$$

$$A = B \sin x = 25.3955 \times 0.02154860 = 0.54723747$$

$$C = \sqrt{0.72^2 - 0.54724^2} = 0.46790$$

$$\frac{C + 0.5\,d_n}{d_n} = \frac{1.1879}{1.44} = 0.825 \text{ flattening}$$

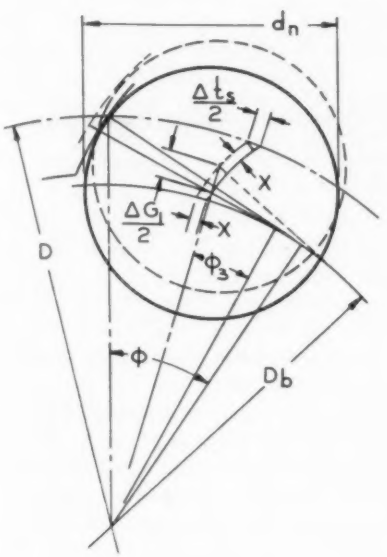
The finest pitch has a greater tolerance, in proportion to size, than the coarsest, so we must check the 48/96 DP. For the 50-tooth internal, then:

$$\text{Cos } x = \frac{(0.5295)^2 + (0.52005)^2 - (0.015)^2}{2 \times 0.5295 \times 0.52005} = 0.999753601$$

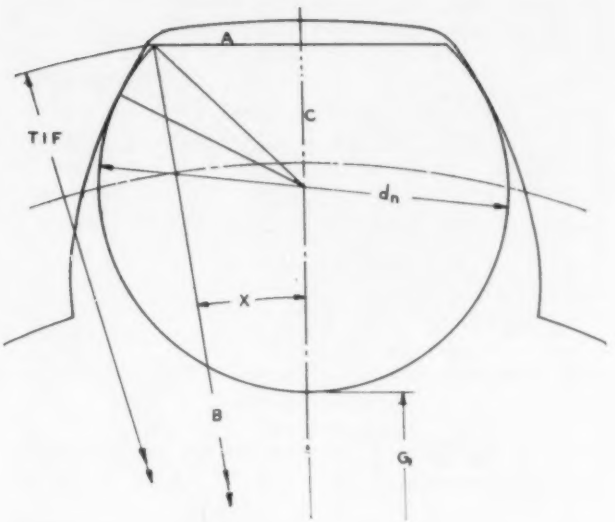
$$A = B \sin x = 0.5295 \times 0.02219757 = 0.011753613$$

$$C = \sqrt{0.015^2 - 0.0117536^2} = 0.0093$$

$$\frac{C + 0.5\,d_n}{d_n} = \frac{0.0093 + 0.015}{0.030} = 0.81 \text{ flattening}$$



■ Fig. 16 – Original position of pin shown as solid line; secondary position shown as dotted line – internal spline [Fig. 11]



■ Fig. 17 – 1.44 pin flattened sufficiently to prevent corner touching involute or fillet under extreme conditions

Another check of the required flattening is to obtain the difference between the minimum major diameter and the sum of the maximum pin measurement and twice the pin diameter. This obtains the interference at the center only, and therefore is not accurate. A check of the same items as above shows the values given in Table 3.

Table 3 – Interference

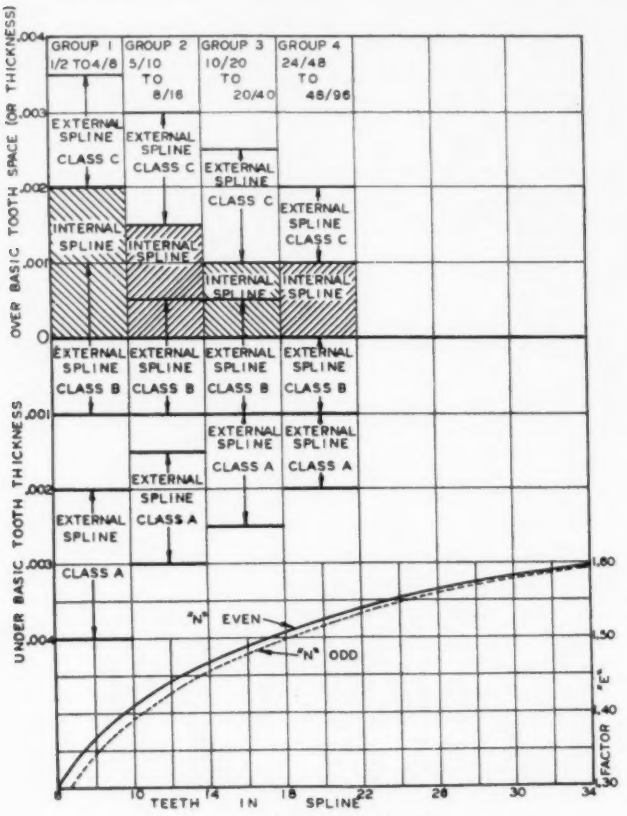
DP	N	G + Tolerance + 2 d_n	D_o Min	Interference	Interference + Fillet Height	Flat
1/2	6	7.2771	7.000	0.1385	0.1905	0.867
1/2	50	51.2834	51.000	0.1417	0.2487	0.828
48/96	6	0.0998	0.1458	0	0	0
48/96	50	1.0701	1.0625	0.0038	0.0058	0.807

A comparison of the above check results with the calculated ones shows a very close agreement for 50 teeth, but for 6 teeth the greater allowance for curvature of the major diameter accounts for the larger difference.

In the standard, a flattening of 0.80 is chosen because it provides for the extreme. Because of the danger of turning the pin so that the diameter loses contact with the involute, it is better practice, where many measurements are taken, to calculate the proper flattening. This will result in longer life of the measuring pin as more floating of the pin may be allowed.

Fig. 14 shows that with the 0.80 flat there is an excess of clearance if the pin used is larger than 1.44; it also shows the possibility of using a special size pin without any flattening. However, as the pin contact approaches the minor diameter of the spline, the angle between the radius to the tangent point and the center of tooth space approaches 90 deg, or the wedging angle approaches O. When this happens, there is an excess of pin measurement variation for even small increments of tooth thickness, and the tolerances are hard to hold. It is, therefore, desirable to choose a pin which will contact at or near the pitch line.

For the full fillet type of spline, the 1.44 pins may be



■ Fig. 18 – Tooth side fits

used without flattening, as the greater major diameters allow ample room for pin clearance.

■ Three-Pin Method

Another method of checking with pins is to use three pins in combination with an arbor whose diameter is just tangent to the pins. This is really a step toward the plug gage, without going to the expense of making a complete gage, but does usually require a special arbor. Its advantage is that the same data may be used for odd or even numbers of teeth, the shaft diameter being always:

$$G_1 = \frac{d_b}{\cos \phi_s} - d_n$$

■ Variations

If two mating splined parts are cut with a circular tooth clearance of 0.001, and one or several teeth are out of correct spacing in the same direction 0.0005, there will still be 0.0005 clearance. If some teeth are out 0.0005 in one direction, and some others are out 0.0005 in the opposite direction, there will be no clearance left. If the total variation from plus to minus is greater than the desired clearance, there will be an interference which will still further be aggravated by any tooth form outside the true involute. The plug gage takes these elements into account, so that any part which accepts a "go" gage will assemble with its mating part as designed.

The Vinco Corp. has contributed materially to the art of spline production. Practical experience with splined fittings frequently shows a discrepancy between actual tooth thickness and space width dimensions (as measured with pins or tooth calipers) and the backlash when the parts are assembled. It appears that the fit of two mating parts is usually tighter than anticipated from the difference between the space width of the splined hole and the tooth thickness of the splined shaft. The reverse condition occurs when parts are checked by means of spline gages having a full complement of teeth. In this case, the part splines appear to be looser in the assembly.

These inconsistencies frequently cause serious difficulties in the production and inspection of splined parts. They are caused by spacing errors and other unavoidable deviations. The following concept constitutes an attempt to clarify these conditions by proposing a new terminology intended to simplify the consideration of these complex conditions. There is every reason to expect that the adoption of this new terminology will help to eliminate practically all difficulties in the interpretation of inspection results.

■ Interference Error

There are basically four errors which prevent splined shafts of a given tooth thickness fitting splined holes of equal space width, as measured with pins, tooth calipers, or similar means. These errors are:

(a) Spacing errors.

(b) Positive profile errors, that is, profile deviations in the direction of the space.

(c) Alignment errors.

(d) Roundness errors.

Inasmuch as these errors cause interference with the ideal perfect spline contour having the given tooth thickness, their combined effect is designated as "interference error."

■ Effective Tooth Thickness and Space Width

If a splined shaft has a measured tooth thickness t and its interference error amounts to e, its effective contour will partly interfere with the ideal contour of the tooth of thickness t. Points on the actual contour will be outside the perfect contour by as much as e inches. The fit of this splined shaft, or its backlash with a mating member, will be the same as if it had a tooth thickness equal to t plus e. This corrected tooth thickness is termed "effective tooth thickness," since it expresses the effect of all irregularities on the fit obtained in the assembly. The tooth thickness t as measured is designated as the "actual tooth thickness" of the splined shaft.

The effective tooth thickness equals the actual tooth thickness plus the interference error.

The same reasoning applied to splined holes leads to the terms "effective space width" and "actual space width."

The effective space width equals the actual space width minus the interference error.

When two splined parts are assembled, the fit between both members depends only on the effective space width, of the splined hole and the effective tooth thickness of the splined shaft. If these two dimensions are equal, a metal-to-metal fit, although only at spots, should be expected even if the actual space width of the splined hole is larger than the actual tooth thickness of the splined shaft. An excess of effective space width over the effective tooth thickness of the mating member results in backlash, while the reverse condition produces press fit.

Backlash equals effective space width of splined hole minus effective tooth thickness of splined shaft.

The difference between the effective and actual dimension on each member is a measure of the uniformity of the spline and has a definite relation to the wear life of the splined fitting. An excessive interference error causes loss of spline contact and excessive stress in the few contacting spline teeth.

■ Part Specifications

The terminology presented above requires new methods of specifying part splines. It must be recognized that each manufactured splined shaft has an actual tooth thickness and an effective tooth thickness and that each manufactured splined hole has an effective space width and an actual space width. If all spline elements were measured on each part coming from a production line, it would be possible to establish:

Maximum effective tooth thickness.
Minimum effective tooth thickness.
Maximum actual tooth thickness.
Minimum actual tooth thickness.
Maximum effective space width.
Minimum effective space width.
Maximum actual space width.
Minimum actual space width.

In determining the specifications for each member of the splined fitting, it will be necessary to decide which of the dimensions listed above should be controlled.

In considering a splined shaft, it will always be necessary to establish a definite limit for the maximum effective tooth thickness which controls the tightest fit.

The maximum actual tooth thickness need not be less and cannot be more than the maximum effective tooth thickness. Yet, its specification may be desirable if the maximum interference error produced by the available machine tool equipment is known. The specified maximum actual tooth thickness provides a basis to check the shaft spline while it is still in the machine and thus make sure that the finished part will have an effective tooth thickness which does not exceed the specified maximum. On the other hand, if difficulties arise in the final inspection of the part spline, the specification of a maximum actual tooth thickness will make it possible to determine whether the interference error or the tooth thickness is excessive.

The minimum actual tooth thickness should always be specified in order to make sure that the backlash will not exceed its absolute maximum, even after the high spots of the spline profile are partly or completely removed by wear or deflection.

The minimum effective tooth thickness cannot be less and need not be more than the minimum actual tooth thickness. It might, therefore, be only of minor importance, unless no great wear or deflection is expected.

(a) If the minimum effective tooth thickness is not specified and the maximum actual tooth thickness is also eliminated, there will be only two limits on the part print, the maximum effective and the minimum actual tooth thickness. In this case, all errors are to be absorbed within the given limits. It will be necessary, however, in establishing these limits to make sure that such absorption is possible.

(b) If a maximum effective tooth thickness is specified in conjunction with the maximum actual tooth thickness, the difference between these two specifications may be determined on the basis of the maximum anticipated interference error.

(c) It is also possible to establish effective tooth thickness limits which allow for only partial accumulation of interference errors. In this case, the difference between the specified effective and actual dimension represents only a percentage of the total interference error.

In expanding the concept of this paragraph to apply to splined shafts, and also to splined holes, the following general conclusions may be established in the form of three possible alternatives for the spline designer:

A. All interference errors are to be absorbed within the given tooth thickness, or space width limits.

B. All interference errors are permissible in addition to the tolerance on the actual tooth thickness or space width.

C. Only a portion of the interference errors is permissible in addition to the tolerance on actual tooth thickness or space width.

If the amount to which interference errors are allowed to be accumulated is designated as "interference allowance," it will be zero in (A), equal to the total interference error in (B), and equivalent to part of the interference error in (C).

The proper selection of interference allowance depends on the evaluation of the interference error as produced by the available tool equipment, on the design requirements, and the inspection procedure. The requirements consist of so many factors that a general requirement cannot be formulated.

■ Side Bearing Fits

Because of the extensive use of broaches for cutting the internal spline, and due to the cost of such equipment and its fixed nature, the internal spline is held to the basic dimension with a plus tolerance, just as in screw thread practice the tapped hole is held. The external spline is then made with a high limit on tooth thickness to the desired maximum press or minimum clearance and a minus tolerance applied.

Three classes of fits are provided, within each of which some selection can be exercised. Three divisions in the range of pitches are used, because coarse pitches cannot and need not be held to such close limits as the fine pitches. Table 4 shows the range of clearance or interference for each class. See Fig. 18.

Table 4 – Range of Clearance or Interference

Diametral Pitch	Clearance (+)	Interference (−)	
	Class A	Class B	Class C
1/2 to 4/8	+0.0020 to +0.0060	+0.0030 to −0.0010	0 to −0.0035
5/10 to 8/16	+0.0015 to +0.0045	+0.0025 to −0.0005	0 to −0.0030
10/20 to 20/40	+0.0010 to +0.0035	+0.0020 to −0.0005	0 to −0.0025
24/48 to 48/96	+0.0010 to +0.0030	+0.0020 to 0	0 to −0.0020

The tolerances in each of the three classes for space width and tooth circular thickness are given in Table 5.

The tolerances in the tables refer to tooth thickness, and these must now be converted to measurements over or between pins. The correct method is to use, for each ex-

Table 5 – Tolerances for Space Width and Tooth Circular Thickness

Diametral Pitch	Internal	External	
		Class A	Classes B and C
1/2 to 4/8	+0.0020 to −0.0000	+0.0000 to −0.0020	+0.0000 to −0.0015
5/10 to 8/16	+0.0015 to −0.0000	+0.0000 to −0.0015	+0.0000 to −0.0015
10/20 to 20/40	+0.0010 to −0.0000	+0.0000 to −0.0015	+0.0000 to −0.0015
24/48 to 48/96	+0.0010 to −0.0000	+0.0000 to −0.0010	+0.0000 to −0.0010

treme of tooth thickness, the general formula for the external spline:

$$\text{INV } \phi_2 = \frac{t}{D} + \text{INV } \phi + \frac{d_x}{D \cos \phi} - \frac{\pi}{N} \qquad (1.0)$$

using the minimum dimension of t for the first calculation, and the maximum for the second. It has been found that this method can be short cut to a commercially close approximation. Tables of basic dimensions (where $t = 0.5p$) are provided with a factor for each number of teeth, and these factors are multiplied by the actual difference between basic and desired tooth thickness, and the result added to or subtracted from the basic dimension to obtain the measurements over pins.

The factor E is obtained as shown in Fig. 15 and is $\frac{\cos \phi}{\sin \phi_2}$; it varies from 1.31 for 6 teeth to 1.63 for 50 teeth. A factor F for the internal spline is similarly $\frac{\cos \phi}{\sin \phi_3}$ and varies from 1.91 for 6 teeth to 1.75 for 50 teeth. Since the internal is kept as a standard, there is only rare need of this factor (Fig. 16).

In the standard, tables are given showing the dimensions between pins for internal splines and for all three classes of fits for the external spline. Where fits other than those given are necessary, the method given above may be used to obtain the dimensions.

When special loose fits are required, it is necessary to make several adjustments in dimensions. The extra looseness at the tooth will allow some movement out of axis of the external in relation to the internal, but not by the full amount of the extra clearance, due to spacing errors. An extra amount of tooth side clearance would allow one-half of it in each direction from center, or a total adjustment of the amount of clearance. This assumes no error in spacing or form. If we make an interference allowance for each member of 0.0005, or a total of 0.001, we will then subtract this from the total maximum clearance, and this becomes the amount for adjustment of either TIF of major or minor diameter. Since the internal is standard, the best course to follow is to decrease the major diameter of the external and increase the minor diameter of the internal by this amount. The second choice is to increase the TIF of the internal, and decrease the TIF ot the external by this amount, and of course this would have to be done for a major diameter fit.

■ Special Combinations

While the standard lists only the usual combinations, as 5/10, 10/20, and so on, it is possible, where special reasons exist, to make other combinations. If tooth height is not important, any addendum or dedendum may be chosen to go with the selected number of teeth and diametral pitch. For example, if 5 DP, 10 teeth are desired, the addendum may be that of the 10/20, in which case the spline could be identified as a 5/20. In such a case, the major diameter, minor diameter, and TIF would be taken from the 10/20 table for 20 teeth because this is the pitch diameter of the 5 DP, 10-tooth spline. The pitch diameter, base circle, pin size, and measurement with pins would come from the 5/10 DP tables because these dimensions are all controlled by the 5 DP proportions. In the same way, any two diametral pitch combinations which have identical pitch diameters may be chosen. With any such combination special tools must be obtained, and special pins must also be used. An extra amount ground off the flattened pin might suffice in some cases, but, due to the very short length of involute curve, it is more likely that special pins would be needed to take advantage of the available space.

With any special pitch combinations, the torque capacity will be different from the standard, and formula (1.7) must be used, with due regard to the proper adjustment for height of teeth in contact.

■ Manufacturing Economics

For high production, where machine time plays such an important part, certain factors should be considered. The greater number of splines to be cut, the longer the machine time will be, therefore, the least number of splines which will carry the load should be chosen. Also, two or three tooth hobs will work faster than single tooth hobs but will not be so accurate because of the difficulty of making the hobs accurately. A hunting action between hob and work is also desirable, to distribute the inaccuracies better among all the teeth. For any machine there is a minimum time required to remove one part and set another in place. If an operator is running two machines, the one is working during the time the other is stopped for a change of parts. In such a case there is no economy in having the working machine stop and wait for a change. Therefore, for every combination there will be a minimum change-over time, below which any reduction of machine time will accomplish no saving.

For these reasons and many others it is advisable, before detail drawings are released, to consult the tool engineer, the production expert, the inspector and the metallurgist. No design engineer can hope to keep abreast of every phase of this problem.

The data given throughout the paper has been made available by the subcommittee, and without their assistance this paper could not have been compiled. This committee, made up of: C. H. Stanard, Buick Motor Division; G. L. McCain, Chrysler Corp.; C. R. Staub, Michigan Tool Co.; S. O. Bjornberg, Illinois Tool Works; J. P. Breuer, Barber-Colman Co.; N. Finkelstein, Buick Motor Division; H. H. Gotberg, Colonial Broach Co.; A. E. Leach, Pontiac Motor Division; and C. A. Young, National Broach & Machinery Co., has been working together for nearly four years, and during this time we have accepted and acknowledged gratefully the able assistance of: A. H. Candee, Gleason Works; H. L. VanKeuren, VanKeuren Co.; C. H. Havill, Bendix Aviation Corp.; L. D. Martin, Eastman Kodak Co.; W. L. Barth, General Motors Corp.; H. A. Marchant, Chrysler Corp.; R. S. Burnett, SAE; S. L. Terry, Chrysler Corp.; A. Beam, Vinco Corp., and many others who have commented, favorably and otherwise.

Development and Application of HEATED WINGS

by CAPT. MYRON TRIBUS*
Army Air Forces

THIS report, a discussion of the design problems in heated surface anti-icing equipment, consolidates and compiles all of the heat transfer data known to be available from past experimentation and considered to be required for current and future designs.

Consequently, the author believes that the discussion contained herein will be of assistance in some degree to designers and engineers confronted with problems relating to heated surface anti-icing.

The report deals with a rapid means of calculating heated wing requirements, charts of heat transfer coefficients, a discussion of instrumentation techniques, and a method of calculating surface temperatures in dry air.

THE AUTHOR: CAPT. MYRON TRIBUS' contributions to the solution of the aircraft de-icing problem in the form of the paper appearing in these pages won for him the 1945 Wright Brothers Medal. Capt. Tribus, who entered the U. S. Army Air Forces in 1942 upon graduating from the University of California, prepared this paper while assigned to reducing icing hazards for aircraft at the Air Technical Service Command, Wright Field. He has established a long record for winning awards, having received appointments as La-Verne Noyes Scholar for 1939-1940, as Otto Henry Greenwald Scholar for 1940-1941, and as Isias W. Hellman Scholar for 1941-1942. He has now left military service, and is doing graduate work at his alma mater.

THE conventional heated wing system may be described generally as shown in Fig. 1, which represents the most successful anti-icing system flown by the Air Forces at the time of this writing.

The design of the various portions of the system may be subjected to fairly exact calculations if the leading edge and the heat exchangers are considered to be simple exchangers obeying the usual laws of heat transfer. The airflows and pressure drops in the ductwork may be calculated by those methods accepted as workable and accurate by the heating and ventilating industry.

Inasmuch as the design demands upon the heat exchangers and ductwork depend upon the requirements of the leading edge, it shall be considered first.

Leading Edge Structure

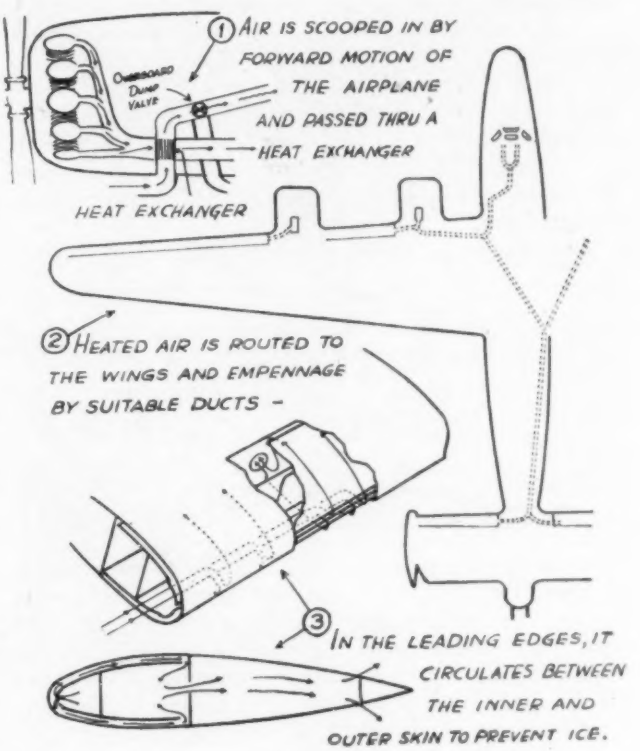
It is necessary that the heated air velocity across the inside surfaces be maintained above a minimum value in order to assure the required heat transfer. The usual method for accomplishing this is to confine the air between the leading edge and an inner skin. A profile view of such a structure is given in Fig. 2.

The plenum (Fig. 2) distributes the heated air spanwise, while the cordwise distribution is as shown. Since the presence of wing tanks makes it impractical to route heated air over the tanks, the air, in such cases, is expelled through louvers in the wing forward of the wing tanks or carried spanwise to a region where it may be conveniently expelled.

Many types of double-skin structure have been proposed. Four examples are shown in cross spanwise section in Fig. 3.

Perhaps the first problem the designer must solve is the extent of the double-skin structure. The theory of ice formation has not progressed sufficiently to allow an exact answer to this question. However, the answer appears to involve the following reasoning:

1. Ice begins to form near the stagnation point. In fact, a narrow region on either side of the stagnation point serves as the primary water-catching surface.


1 AIR IS SCOOPED IN BY FORWARD MOTION OF THE AIRPLANE AND PASSED THRU A HEAT EXCHANGER
OVERCOOLED DUMP VALVE
HEAT EXCHANGER
2 HEATED AIR IS ROUTED TO THE WINGS AND EMPENNAGE BY SUITABLE DUCTS -
3 IN THE LEADING EDGES, IT CIRCULATES BETWEEN THE INNER AND OUTER SKIN TO PREVENT ICE.

■ Fig. 1 – Conventional heated wing system

[This paper was presented at the SAE War Engineering – Annual Meeting, Detroit, Jan. 11, 1945.]

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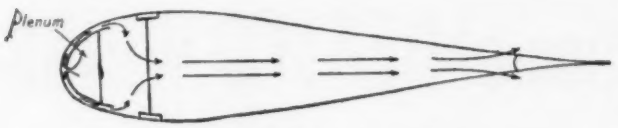


■ Fig. 2 – Method of heating leading edge

2. The formation of ice on portions of the wing other than the stagnation region depends on the amount of runback that occurs. Runback may be described as the phenomenon of freezing of droplets in a region on the wing surface back of the region of primary water catching. This phenomenon is illustrated in Fig. 4.

As shown in Fig. 4, it appears that the proper criterion for the successful prevention of ice is the total heat loss from the leading edge. The smaller the heated area, the greater the intensity of heating required.

At present the only experimental evidence concerning the extent of the required heated area that can be presented is set forth in Table 1, which gives the protected areas and heat losses for three airplanes successfully flown during icing conditions at Minneapolis during the winter of 1942-1943.

Although the protection afforded the B-24 airplane was superior to the protection afforded by the conventional inflatable shoe system, there were a few flights in which small amounts of ice did accumulate on the lesser heated portions of the wings and empennage. These were very minor accretions and did not detract from the flight personnel's enthusiasm for the new system. It is felt that these minor failures were due to insufficient heating, and for this reason production versions of this airplane will be equipped with heat exchangers of an increased capacity.

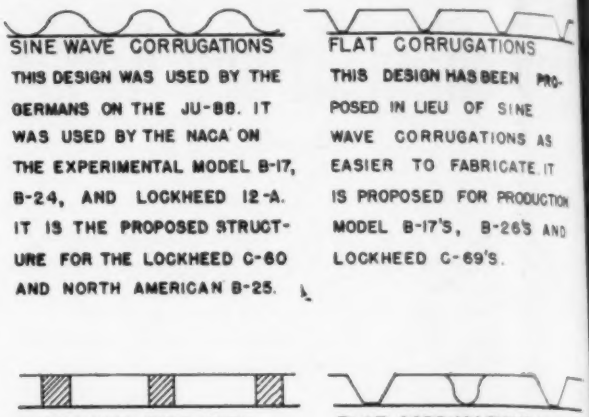
Theoretical calculations indicate more heat is required than supplied in either the B-17 or B-24. In view of this fact, the AAF has set 1500 Btu per hr per sq ft for the forward 15% of chord as the design condition. This figure is a result of a study of the heat transfer coefficients over various airfoils and the probable range of icing conditions to be encountered. It is considered, therefore, a safe design figure.

In this regard, many wind tunnel tests and dry air flight

Table 1 – Heat Losses from Airplanes Successfully Flown in Icing Conditions at Minneapolis during Winter of 1942-1943

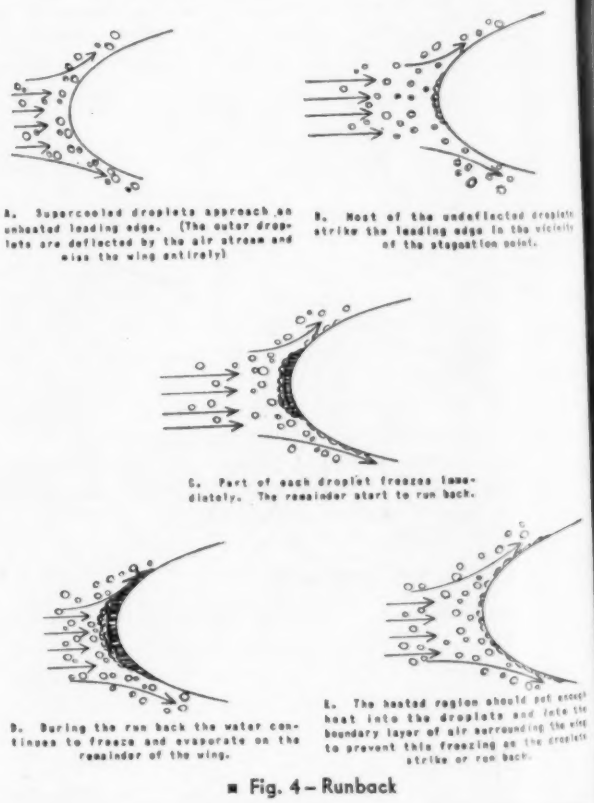
	Protected Area, sq ft	Total Heat Loss, Btu per hr	Average Heat Loss, Btu per hr per sq ft	Part of Chord Directly Heated, %
B-17	68	85,000	1250	14
B-24	45.3	43,000	950	8.1
Lockheed 12A	18.2	45,000	2360	11

tests using water sprays indicate that much greater heat is required than that which has been supplied in successful flights. It is probable that the wind tunnel tests have been overly severe. Nature is particularly generous in that she does not permit large quantities of water to be suspended in the atmosphere. Measurements of fogs at 40 F indicate that 0.5 grains of suspended moisture per cubic foot of air represent a pea-soup fog – one which would represent a fairly severe icing condition were it 10 F colder. It is quite difficult to inject this small amount of moisture into the


SINE WAVE CORRUGATIONS
THIS DESIGN WAS USED BY THE GERMANS ON THE JU-88. IT WAS USED BY THE NACA ON THE EXPERIMENTAL MODEL B-17, B-24, AND LOCKHEED 12-A. IT IS THE PROPOSED STRUCTURE FOR THE LOCKHEED C-60 AND NORTH AMERICAN B-25.
FLAT CORRUGATIONS
THIS DESIGN HAS BEEN PROPOSED IN LIEU OF SINE WAVE CORRUGATIONS AS EASIER TO FABRICATE. IT IS PROPOSED FOR PRODUCTION MODEL B-17'S, B-26'S AND LOCKHEED C-69'S.
FLAT INNER SKIN WITH SPACERS
THIS DESIGN HAS BEEN PROPOSED FOR FAIRCHILD'S XC-82.
FLAT CORRUGATIONS WITH DIMPLES ON INNER SKIN

■ Fig. 3 – Sample double-skin structures

air below freezing temperatures and not have it freeze almost immediately. May it be noted that it takes four times as much heat to change a 50% mixture of ice and water at 32 F to water at 32 F as it does to bring supercooled water from 14 F to 32 F (—10 C to 0 C). Furthermore, unless tests are run at or near 100% relative humidity, accelerated cooling by evaporation is likely to occur. Wind tunnel data and dry air spray tests should be re-evaluated in the light of these factors.


A. Supercooled droplets approach an unheated leading edge. (The outer droplets are deflected by the air stream and miss the wing entirely)
B. Most of the undeflected droplets strike the leading edge in the vicinity of the stagnation point.
C. Part of each droplet freezes immediately. The remainder start to run back.
D. During the run back the water continues to freeze and evaporate on the remainder of the wing.
E. The heated region should put enough heat into the droplets and into the boundary layer of air surrounding the wing to prevent this freezing as the droplets strike or run back.

■ Fig. 4 – Runback

From time to time there has arisen a question concerning afterfreezing on the ailerons due to excessive runback. In the winter's operations in Minneapolis, the problem did not arise on any of the airplanes tested. It is not apparent why such should have been the case, since it has been generally observed that during a rain, water will run back and over the trailing edges. However, it seems that during icing conditions there is not nearly as much water present as in a rain; and, furthermore, with heated leading edges a large quantity of the water appears to evaporate in the leading edge region. Finally, a contributing factor may be that the wing is surrounded by a blanket of warm air which prevents the freezing of the water while traversing the after portions of the wing. Whatever the reasons, the fact remains that afterfreezing did not occur, even in freezing rains.

A second criterion for the design is that the temperature of the airfoil surfaces shall not fall below 32 F at any time. Predicting this temperature for an airfoil during an icing storm is at present an impossible task. Too little is known about the behavior of boundary layers about airfoils in air which is not only highly turbulent but which contains water droplets splattering over and roughening the leading edge surface. For airfoils in dry air some reasonable approximations can be made. An analysis of the heat transfer in the B-17 airplane is compared to flight data in Appendix I.

Heat Exchangers

The greatest difficulty in the design of heated wings has been the fabrication of the proper heat exchanger. On many aircraft the AAF has required that the heat exchangers have adequate capacity to cool the exhaust gases below flaming temperature to prevent afterburning. This latter requirement has generally proved greater than the heated wing requirements, a fortunate fact for the heated wing designer since it assures him of an adequate supply of heated air.

Early designs of heat exchangers attempted to utilize the high thermal conductivity of copper and the corrosive resistance and strength of stainless steel by combining the two metals. Copper fins imbedded in stainless steel casings were used as one form of design. The number of failures and difficulties of manufacture encountered in such a design have caused manufacturers to turn to simpler types. At present, efforts seem to center around all prime surface exchangers as contrasted to exchangers utilizing fins. Some sample designs are shown in Fig. 5.

If the heat exchanger is required for anti-icing only, a saving in weight can usually be made by properly combining the thermal efficiency of the leading edge with that of the heat exchanger. In general, the less air required, the lower the drag on the airplane and the smaller and lighter the heat exchanger.

A quick rule of thumb for determining the size of heat exchanger required is:

$$Q = 875A$$

where:

Q = Heat exchange output, Btu per hr

A = Lift area protected, sq ft

This equation is quite approximate but will serve as a guide to the designer in his initial approach to the problem. The derivation of this equation may be found in Appendix II.

At present there has not been sufficient life testing of the various designs to permit standardization on any one design. Efforts in this direction will be made and it is hoped that eventually one or two designs of heat exchanger core can be made to fit all A N B aircraft.

A tentative standard of 10,000 Btu per hr per lb of exchanger has been set forth by the NACA and appears to be a fairly good design standard. The pressure drop through the exchanger on the exhaust side should not exceed 2 in. of mercury at take-off power. The allowable pressure drop on the air side will be set by the design of the remainder of the wing system. If it is above 6 in. of water it will generally be in the critical range for maintaining proper circulation within the wing.

The design condition for the heat exchanger performance should be at the low-power cruise condition of the airplane. This allows a factor of safety in that when more power is used more heat will be supplied to the wings. In this regard, the data supplied by heat exchanger manufacturers should be carefully examined to ascertain that the rating given a particular heat exchanger can be correctly applied to a given installation. For example, a 10% increase in both air and exhaust rates, along with an increase in exhaust gas temperature of from 1350 F to 1600 F, will show increased outputs of 20 to 30% over what will occur in the actual installation. Heat exchanger manufacturers should be requested to furnish a description of their test technique with the data showing such things as heat balances, probable accuracy of the readings, and

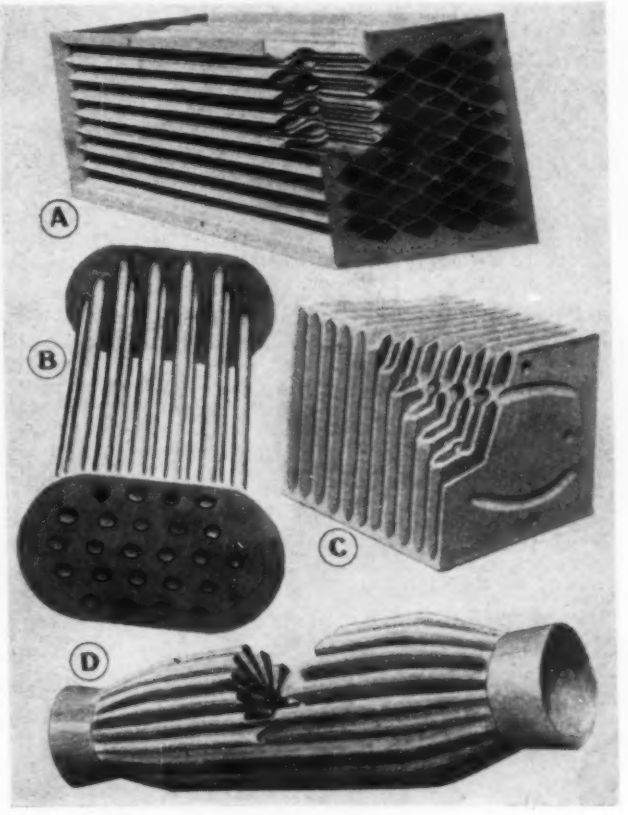
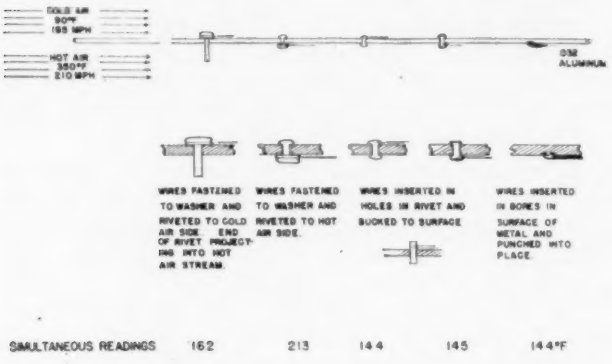
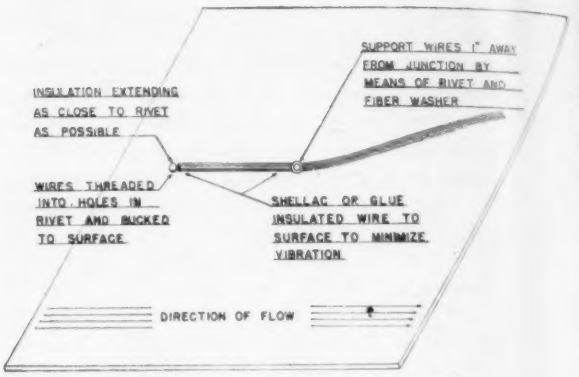


Fig. 5 – Types of heat exchanger: *A* formed plate; *B* tube bundle; *C* flat plate; and *D* hollow fin



■ Fig. 6 – Comparative tests of thermocouple installations


SUPPORT WIRES 1" AWAY FROM JUNCTION BY MEANS OF RIVET AND FIBER WASHER
INSULATION EXTENDING AS CLOSE TO RIVET AS POSSIBLE
WIRES THREADED INTO HOLES IN RIVET AND BUCKED TO SURFACE
SHELLAC OR GLUE INSULATED WIRE TO SURFACE TO MINIMIZE VIBRATION
DIRECTION OF FLOW

■ Fig. 7 – Thermocouple installation for measuring skin temperatures

whether shielded or unshielded thermocouples were used in measuring exhaust gas temperatures.

■ Ductwork

The calculation of airflows within the ductwork follows the methods usually employed in treating industrial installations. A convenient compilation of friction factors and shock loss coefficients is given in an SAE Aeronautical Information Report.[1] The methods of calculation presented therein are recommended for use in heated wing systems.

The pressure drops in the wing leading edge require some attention in order to assure a reasonable air distribution. The plenum should be adjusted spanwise to account for the loss of air through each corrugation. On most aircraft with tapered wings it will be found that a duct which decreases in cross-section linearly with span will be satisfactory.

■ Instrumentation

The instrumentation in the heated wing system should supply sufficient information for the design engineer to check the assumptions upon which his calculations were based. This will include air and exhaust gas temperatures and pressures, as well as structural temperatures. Wherever possible, venturi meters or orifices should be inserted in the ducts to measure airflow.

Thermocouples used to measure exhaust gas temperatures should be radiation-shielded.

The method of affixing thermocouples to surfaces warrants considerable attention. False readings can easily be obtained by faulty installations. For example the following test can be cited:[2]

A sheet of 0.032 aluminum was subjected to hot air on one side and cold air on the other. Thermocouples were fastened in different fashions to both sides. The differences in readings were used as a comparison of the errors inherent in each design. The results and designs are shown in Fig. 6.

From these tests it is recommended that an installation similar to that shown in Fig. 7 be used wherever skin temperatures are desired. Iron and constantan wire is at present most commonly used by industry.

■ Miscellaneous Controls, Safety Devices

Military aircraft require extreme simplicity of operation and maintenance for all component parts. This means that many refinements desirable in commercial aircraft must be foregone. Icing indicators, for example, are not desired. Controls which permit the flight engineer to vary the exact heat output of his exchangers are considered as invitations to maintenance troubles. The controls for the heated wing should be a simple "on – off" type. Safety switches to shut off the system in case of overheating should be placed downstream of each exchanger. They should be so installed that the pilot or copilot can override them in an emergency.

On multiengined aircraft it is considered a desirable feature to be able to utilize the anti-icing system even in the event of an engine failure. This feature requires extra ducting and controls to allow the heat from the operative engines to be redistributed. However, the added weight of the cross-over provisions may be prohibitive. For this reason it is the Air Forces policy to decide each case as it arises. In any event, the system should be so designed that it is impossible to heat one wing and not the other.

■ Conclusion

The statements and methods of calculation presented in this report are not to be considered as final in any case. As experience is gained in the use of heated wings, revisions will most certainly be found necessary. The current requirements for the Army Air Forces will be found in Specification R-40395, "Heated Surface Anti-Icing, General Specification for."

Appendix I

Thermal Analysis of the Leading Edge

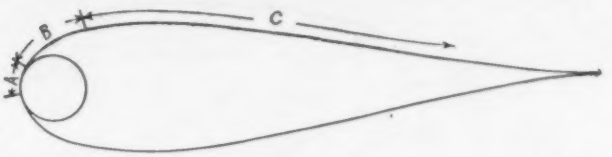
The double-skin region corresponds to a heat exchanger in which the air is cooled. As such, the usual heat exchanger nomenclature and methods of calculation apply. As in any exchanger, it is necessary to know the external and internal heat transfer coefficients involved and the area of heat transfer surface.

1. *The External Heat Transfer Coefficient* – Various

[1] See SAE Aeronautical Information Report No. 2, Jan. 1, 1943, "Airplane Heating and Ventilating Equipment, Engineering Data."

[2] Taken from Douglas Aircraft Report ES-6026, "Development Tests of Various Methods of Heat De-Icing Horizontal Stabilizer."

■ Fig. 8 – Airfoil idealized as cylinder and series of flat plates



■ Fig. 9 – *A* cylinder region; *B* laminar plate region; and *C* turbulent region

methods are available for calculating the heat transfer coefficient over the outside of an airfoil. The simplest method for those not especially trained in aerodynamics is that given by Martinelli.[3] In brief this report idealizes an airfoil as shown in Fig. 8.

The airfoil is considered as a cylinder and a series of flat plates. The heat transfer properties of cylinders and flat plates have been measured in wind tunnels and are fairly well known. It remains, therefore, to set up simple equations to describe the heat transfer coefficient for each part of the airfoil.

The flow about an airfoil can be divided into the three regions, (1) the cylinderlike region near the stagnation point, (2) the laminar flow region where the airfoil contour ceases being a cylinder, and (3) the turbulent flow region, as shown in Fig. 9.

(1) For the cylinderlike region the local heat transfer coefficient is given by Martinelli as (Fig. 10):

$$f = 0.194 T^{0.49} \left(\frac{V\gamma}{d}\right)^{0.5} \left[1 - \left(\frac{4L}{\pi d}\right)^3\right] \qquad \text{(A)}$$

where:

f = Heat transfer coefficient, Btu per hr per sq ft per F

T = Absolute temperature averaged between free air temperature and the surface (460 + T F), R

γ = Air density, lb per cu ft

d = Diameter of "equivalent cylinder," ft

L = Distance from stagnation point, ft

V = Free stream velocity, fps (*not* indicated air speed)

This formula has been well substantiated by wind tunnel data in the referenced report.

(2) For the laminar region Martinelli presents (Fig. 11):

$$f = 0.0562 T^{0.5} \left(\frac{U\gamma}{L}\right)^{0.5} \qquad \text{(B)}$$

where:

f = Heat transfer coefficient, Btu per hr per sq ft per F

T = Absolute temperature averaged between free air temperature and surface, R

U = Local velocity just outside the boundary layer, fps

L = Distance from stagnation point, ft

γ = Air density, lb per cu ft

This equation actually is based upon flat plate experiments and is not strictly applicable to airfoils where pressure gradients may affect the heat transfer. A more complex solution is given by Allen and Look[3] and by Frick and McCullough[3] who write:

$$\frac{f\,\delta_L}{k} = \lambda$$

$$\delta_L = c\sqrt{\frac{5.3}{R_c}\left(\frac{V}{U}\right)^{0.17} \int_0^{L_1/c} \left(\frac{U}{V}\right)^{0.17} d\,\frac{L}{c}}$$

where:

f = Heat transfer coefficient, Btu per hr per sq ft per F

δ_L = Laminar boundary layer thickness, ft

k = Thermal conductivity of air, Btu per hr per sq ft (F per ft)

λ = Experimental constant, dimensionless

c = Chord length, ft

R_c = Reynolds number based on chord length, $\frac{VC\rho}{\mu}$

V = Free stream velocity, fps

ρ = Air density, slugs per cu ft

μ = Absolute viscosity of air, lb-sec per sq ft

U = Local velocity just outside the boundary layer, fps

U_1 = Local velocity at L_1, fps

L = Distance along surface from stagnation point, ft

The above equation can be simplified to:

$$\mathrm{Nu}_x = \lambda \sqrt{\frac{\mathrm{Re}_x}{5.3}} \sqrt{\frac{U_1^{3.17} L_1}{\int_0^{L_1} U^{3.17} dL}}$$

where:

Nu_x = Nusselt's number based on distance from stagnation point, $\frac{fL}{k}$

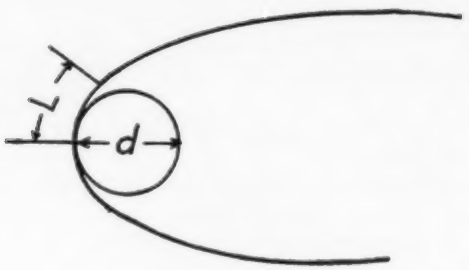
Re_x = Reynolds number based on distance from stagnation point, $\frac{UL\rho}{\mu}$

Allen and Look assign a value of 0.765 to λ, while Frick and McCullough use 0.700. For the flat plate the second square root becomes equal to unity. Three equations are, therefore, given by three sets of authors:

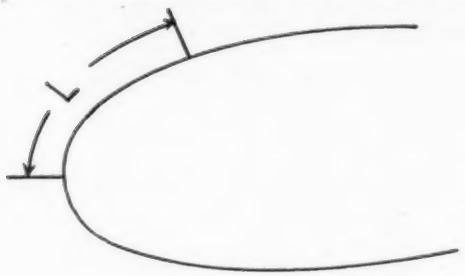
$$\left.\begin{aligned} \mathrm{Nu}_x &= 0.303\sqrt{\mathrm{Re}_x} && \text{Frick and McCullough} \\ \mathrm{Nu}_x &= 0.332\sqrt{\mathrm{Re}_x} && \text{Allen and Look} \\ \mathrm{Nu}_x &= 0.295\sqrt{\mathrm{Re}_x} && \text{Martinelli and others} \end{aligned}\right\} \text{Flat plate}$$

For the airfoil, then, according to Frick and McCul-

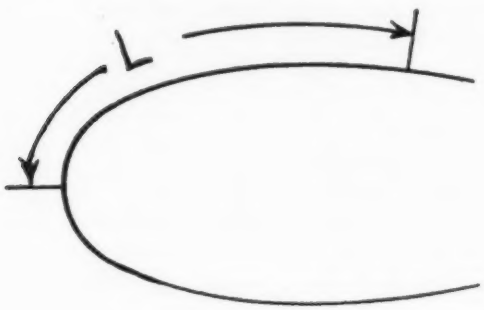
[3] See NACA ACR Report, March, 1943, "Investigation of Aircraft Heaters, Section VIII, Simplified Method for Calculation of Unit Thermal Conductance over Wings," by R. C. Martinelli and others; other reports are: NACA Restricted Bulletin, December, 1942, "Method for Calculating Heat Transfer in Laminar Flow Region of Bodies," by H. J. Allen and B. C. Look; NACA ACR, December, 1942, "Method for Determining Rate of Heat Transfer from Wing or Streamline Body," by C. W. Frick and G. B. McCullough; and RAE Report No. Aero. 1783, November, 1942, "Heat Transfer Calculation for Airfoils," by H. B. Squire.



■ Fig. 10 – Cylinderlike region



■ Fig. 11 – Laminar region



■ Fig. 12 – Turbulent region

lough, or Look and Allen, the flat plate equations must be multiplied by the quantity

$$\sqrt{U_1^{0.17} L_1 \Big/ \int_0^{L_1} U^{0.17} dL}$$

For the region of laminar flow, where the velocity is generally increasing, this factor is greater than unity. For example, where the increase in velocity is linear with L, the correction factor is 3.05. This figure is based upon the assumption that the Blasius type of velocity profile exists in a laminar boundary layer in the stagnation point region, which may or may not be true.

As a matter of practical engineering, however, it will be found very difficult to obtain pressure distribution data which will allow accurate calculation of this correction factor. An error of 5% in the determination of a velocity, for example, yields a 41% error in the product $U^{0.17}L$. Generally, pressure diagrams are given in terms of percentage of chord, which makes the values of the velocity near the stagnation point nearly indeterminate. Also, it is to be remembered that the derivations of Frick and McCullough, or of Look and Allen, apply accurately only to an airfoil which is at uniform temperature over the surface.

For the calculations presented herein the flat plate equations were used in the uncorrected form.

(3) The turbulent region:

For the turbulent region the flat plate equations apply fairly well. For this region Martinelli gives (Fig. 12):

$$f = 0.524 T^{0.296} \left(\frac{U\gamma}{L^{0.25}} \right)^{0.8} \qquad \text{(C)}$$

where:

f = Heat transfer coefficient, Btu per hr per sq ft per F

T = Absolute temperature averaged between free air temperature and surface (460 + F), R

U = Local velocity at L, fps

γ = Air density, lb per cu ft

L = Distance from stagnation point, ft

This checks quite closely with the values calculated by the more complex method of Frick and McCullough and is the approach hinted at by Squires. The three foregoing equations (A), (B), and (C) then will describe the heat transfer coefficient about an airfoil in dry air, provided the pressure distribution and transition points are known. In the absence of better information, it is probably best to assume transition at the minimum pressure point.

2. *The Internal Heat Transfer Coefficient:*

$$f = 5.56 \times 10^{-4} \frac{T^{0.296} W^{0.8}}{D^{0.2} A_e^{0.8}} \qquad \text{(D)}^6$$

where:

f = Heat transfer coefficient, Btu per hr per sq ft per F

T = Average absolute temperature of air flowing, R

W = Weight rate of flow, lb per hr

D = Equivalent diameter of air passage, ft

$$D = 4 \times \frac{\text{Cross-sectional area of flow}}{\text{Wetted perimeter}}, \text{ ft}$$

A_e = Cross-sectional area of air passage, sq ft

The heat transfer coefficient in odd-shaped passages of many types have been calculated successfully by means of the above equation.

As an example of the use of these equations an analysis of the flight test data of a B-17 airplane flown in Minneapolis during the winter will be presented.

The airflow rates in each corrugation for certain design conditions were calculated and presented by Look.[5] The

■ Fig. 13 – Sine-wave corrugation of 1-in. pitch and 1/4-in. depth

[4] See NACA ARR, October, 1942, "Investigation of Aircraft Heaters. Section III, Measured and Predicted Performance of Double Tube Heat Exchangers," by R. C. Martinelli and others.

[5] See NACA ACR, Nov. 7, 1942, "Analysis of Thermal Ice Prevention Equipment for Boeing B-17F Airplane," by B. C. Look and others.

data on airflows presented in the above report will be used since the calculated pressure drops throughout the system closely follow the experimental values.

The corrugations on the B-17 airplane had the dimensions shown in Fig. 13.

The weight rate of flow through the two stations to be considered is given in Table 2.

The internal heat transfer coefficients as a function of weight flow are given in Fig. 14. These coefficients will predict results slightly low, since the sides of the corrugations act somewhat as fins to give increased heat transfer. The greater the pitch of the corrugation, the less this fin effect will be.

The local velocity over the airfoil may be calculated from the pressure distribution over the airfoil. An NACA report by Garrick[6] was used to obtain the pressure distributions over NACA airfoil sections 0010 and 0012 as representative of stations 33 and 25. (Station 33 is actually closer to NACA 0009 section.) These pressure distributions are reproduced in Fig. 15.

The velocity distribution is calculated from the pressure distribution by means of the equation

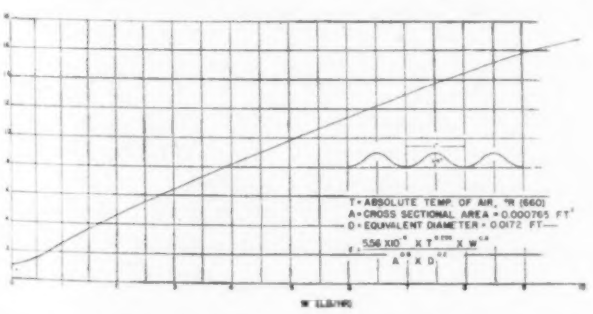
$$\frac{U}{V} = \sqrt{1 - \frac{\Delta P}{q}}, \text{ where } \frac{\Delta P}{q} =$$

ratio of superstream pressure to dynamic pressure. The velocity distribution is given in Fig. 16.

The transition point for airfoils of the NACA 0009 and 0012 sections has been measured and is reported for low velocities by Silverstein and Becker.[7] When these data are extrapolated to the flight conditions of the test, they indicate transition near the 4% of chord line on the upper surface. A value of the transition point of 3½% on the upper surface and 8% on the lower surface was arbitrarily chosen, since no information on the comparative roughness was available.

Table 2 – Weight Rate of Flow through Two Stations

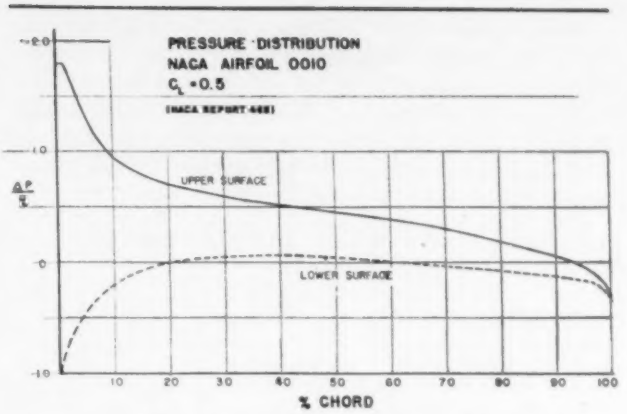
	Design Conditions	Flight 1	Flight 2
Total Flow to Wing, lb of air per hr	2730	2290	1940
Flow at Station 25, lb of air per hr corrugation	6.48	5.40	4.60
Flow at Station 33, lb of air per hr corrugation	7.37	6.15	5.23

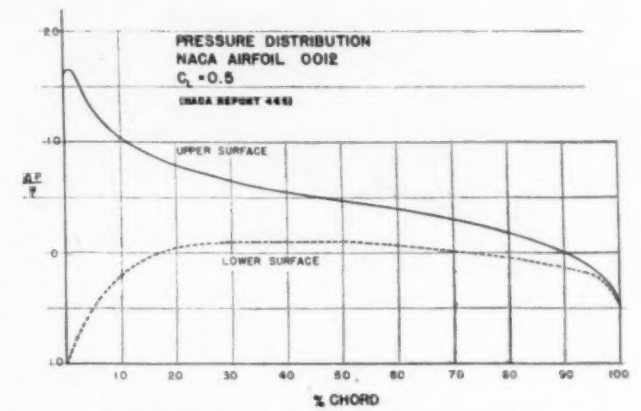


■ Fig. 14 – Internal heat transfer coefficients as function of weight flow

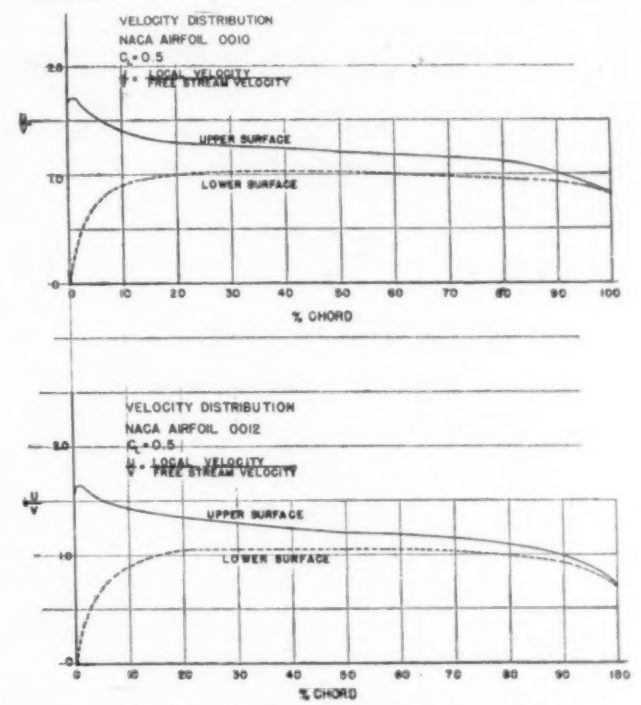
[6] See NACA Technical Report No. 465 (1933), "Determination of Theoretical Pressure Distribution for 20 Airfoils," by I. E. Garrick.

[7] See NACA Technical Report No. 637 (1938), "Determination of Boundary-Layer Transition on Three Symmetrical Airfoils in NACA Full-Scale Wing Tunnel," by A. Silverstein and J. V. Becker.





■ Fig. 15 – Pressure distribution over two airfoils



■ Fig. 16 – Velocity distribution for two airfoils

Using equations (A), (B), and (C), the local heat transfer coefficients were calculated and plotted in Figs. 17 and 18 to represent the two stations during flight Nos. 1 and 2.

The average external heat transfer coefficients are plotted in Figs. 19 and 20. These are average values from the stagnation point back to the percentage of chord under consideration.

The overall heat transfer coefficient is defined as:

$$\frac{1}{U} = \frac{1}{f_i} + \frac{1}{f_o} + \frac{\Delta x}{k}$$

where:

f_i = Internal heat transfer coefficient
f_o = External heat transfer coefficient
Δx = Thickness of metal
k = Thermal conductivity of metal
U = Overall heat transfer coefficient

The heat lost by the air going through the corrugations is equal to that transferred through the outer skin. That is:

$$W \times C_p \times (t_{a_1} - t_{a_2}) = U_{av} \times A \times \theta_{lm} \qquad \text{(E)}$$

where:

W = Weight of air flowing, lb per hr
C_p = Specific heat, Btu per lb per F
$t_{a_1} - t_{a_2}$ = Temperature drop of air, F
U_{av} = Average overall heat transfer coefficient, Btu per hr per sq ft per F
A = Area of heat transfer, sq ft
θ_{lm} = Logarithmic mean temperature difference, F

In the temperature range under consideration, the log mean temperature difference is equal to the arithmetic average of the temperature differences, that is:

$$\theta_{lm} = \frac{t_{a_1} + t_{a_2}}{2} - t_o$$

where:

t_{a_1} = Entering air temperature

t_{a_2} = Exit air temperature

t_o = Outside air temperature

If all temperatures are measured relative to the outside temperature, that is, $t_o = 0$, equation (E) simplifies to:

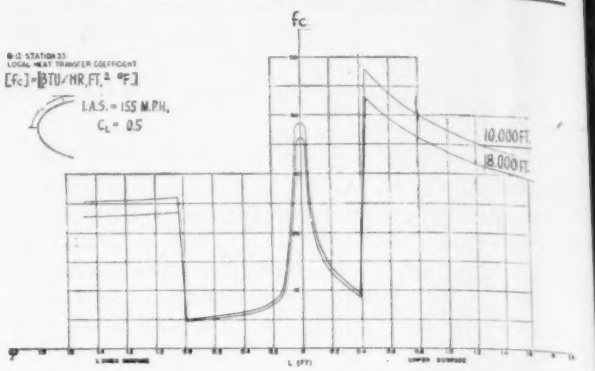
$$\frac{t_{a_1} - t_{a_2}}{t_{a_1}} = \eta = \frac{2}{\frac{2WC_p}{UA} + 1}$$

where:

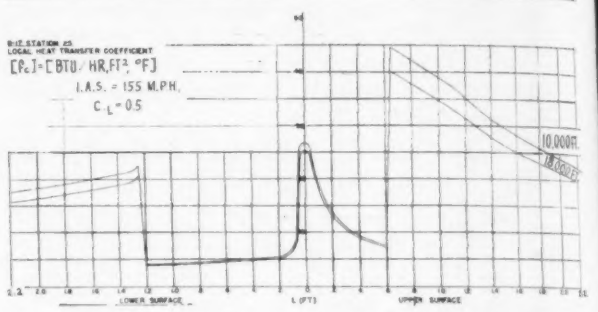
η = Efficiency of heat removal in the double-skin region

In Fig. 21, the calculated values of η are compared to the measured values in flights Nos. 1 and 2.

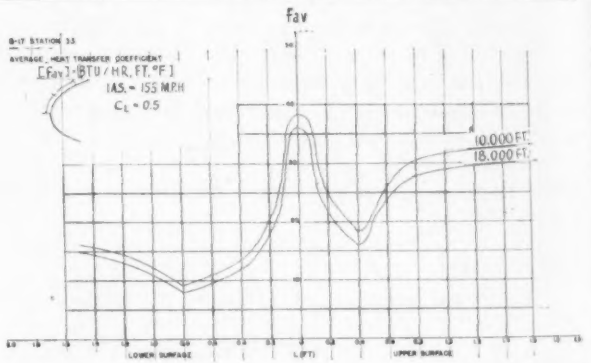
It will be noted that exact knowledge of the external heat transfer coefficients is not necessary for design, as the resistance to heat transfer on the inside is generally the greater of the two. However, calculation of the skin temperatures requires exact knowledge of both internal and external coefficients. The relation between skin temperatures and air temperatures is given by the ratio of the local



■ Fig. 17 – Local heat transfer coefficient at station 33



■ Fig. 18 – Local heat transfer coefficient at station 25



■ Fig. 19 – Average external heat transfer coefficient at station 33

heat transfer coefficients as follows:

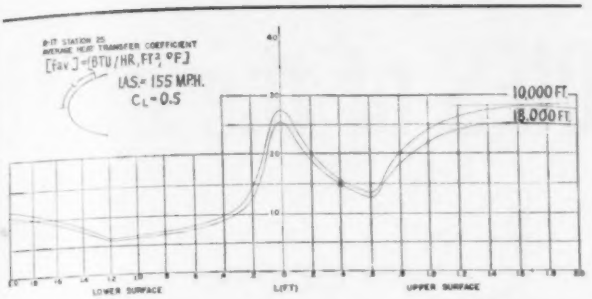
$$\frac{t_{skin}}{t_{air}} = (1 - \eta)\frac{f_i}{f_o + f_i}$$

where:

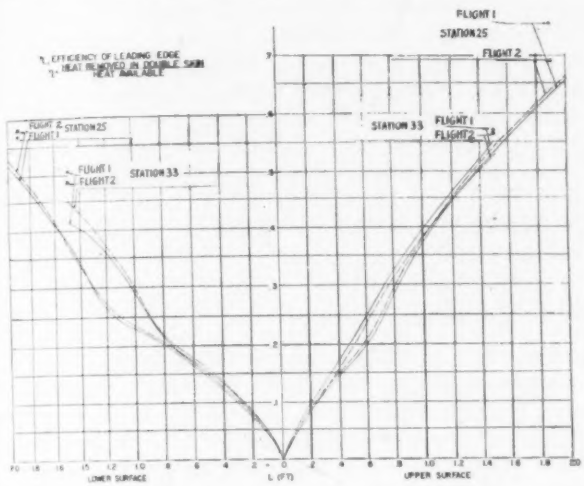
t_{skin} = Skin temperature above free air temperature
t_{air} = Initial temperature use of the air
η = Efficiency of leading edge to point where t_{skin} is desired
f_i = Local external heat transfer coefficient
f_o = Local internal heat transfer coefficient

The calculated skin temperatures are given for stations 33 and 25 in Figs. 22 and 23. It will be noted that generally the predicted temperatures are lower than the measured values.

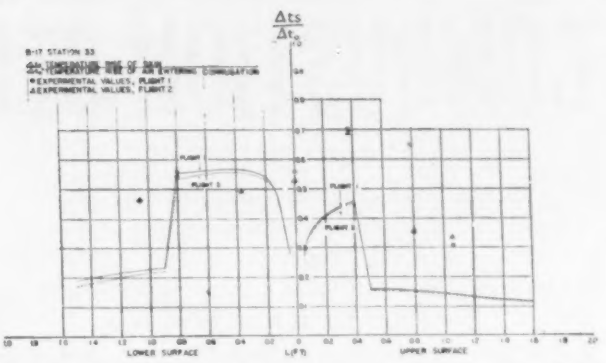
No attempt has been made to predict the temperature rise in icing conditions. The following reasoning, however, may be used to give a clue as to the extra cooling effect of the water. The air-water mixture may be regarded as



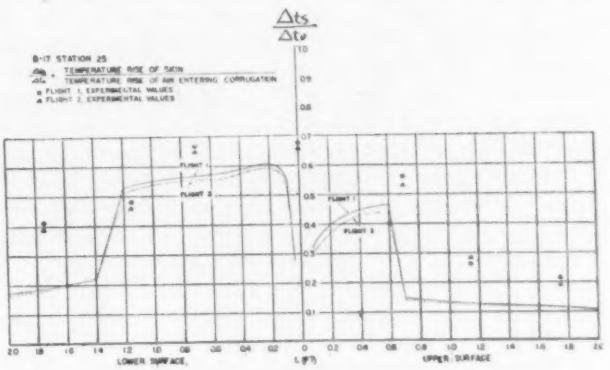
■ Fig. 20 – Average external heat transfer coefficient at station 25



■ Fig. 21 – Efficiency of heat removal in double-skin region



■ Fig. 22 – Calculated and experimental skin temperatures[8] for station 33



■ Fig. 23 – Calculated and experimental skin temperatures[8] for station 25

homogeneous and ascribed to a specific heat, viscosity, and thermal conductivity as with any fluid. In general, the heat transfer coefficient varies with the 1/3 power of the specific heat and 2/3 power of the thermal conductivity. It varies somewhat less with the viscosity. The largest change in properties probably will be in that of the specific heat. At 32 F the increase in the specific heat of saturated air over dry air is 88%. By this reasoning we would expect an increase of about 23% in the external heat transfer coefficient at 32 F. This increase should become less at lower temperatures and vanish around 0 F, since saturated air at this low temperature contains so little water. This is considered fortuitous in that if the wing is designed for low-temperature icing, it will be successful at moderate temperatures, even if the temperature rise is not as great at the higher temperature.

Appendix II

Derivation of Equation for Determining Size of Heat Exchanger

A rough estimate of the heat exchanger output required for a heated wing may be made if the following design conditions are assumed:

1. The temperature rise of the air through the exchanger is 350 F.

[8] Since the preparation of this report information has been received from the NACA that the surface temperatures reported by them for the B-17 are in excess of the true temperatures by 20-30 F. This correction will bring the experimental points in much closer agreement with the calculated curves than shown in Figs. 22 and 23, which are based on the published results.

2. The forward 15% of chord is to be protected.

3. The average rate of heating is 1500 Btu per hr per sq ft over this region.

4. The efficiency of heat removal in the double-skin region is approximately 60%.

The heat lost by the air equals the heat lost through the double skin.

$$Q = WC_p\,(t_{in} - t_{out}) = A \text{ (protected area)} \times 1500$$

$$W \text{ (Weight of air)} = \frac{A \text{ (protected area)} \times 1500}{C_p \text{ (heat capacity of air)}\ (t_{in} - t_{out})\ \text{(temperature drop of air)}}$$

$$(t_i - t_0) = 0.60 \times 350 = 210 \text{ [1 and 4]}$$

$$A = 2 \times \frac{15}{100} \quad A \text{ (lift area)}$$

$$A \text{ (protected)} = 0.3\ A \text{ (lift)}$$

$$W = \frac{1500 \times 0.3\ A \text{ (lift)}}{0.243 \times 210} \text{ lb per hr}$$

$$W \approx 10\ A \text{ (lift) lb per hr}$$

$$Q \text{ (capacity of heat exchangers)} \approx 850\ A \text{ (lift) Btu per hr}$$

Appendix III

Bibliography

1. NACA ACR, April, 1941, "Preliminary Report on Flight Tests on Airplane Having Exhaust-Heated Wings," by L. A. Rodert and others. A report on the design of the exhaust-heated Lockheed 12A. This airplane had an exhaust tube through the leading edge which warmed the surfaces by radiation and convection.

concluded on page 309

COMBUSTION STUDIES of the DIESEL E

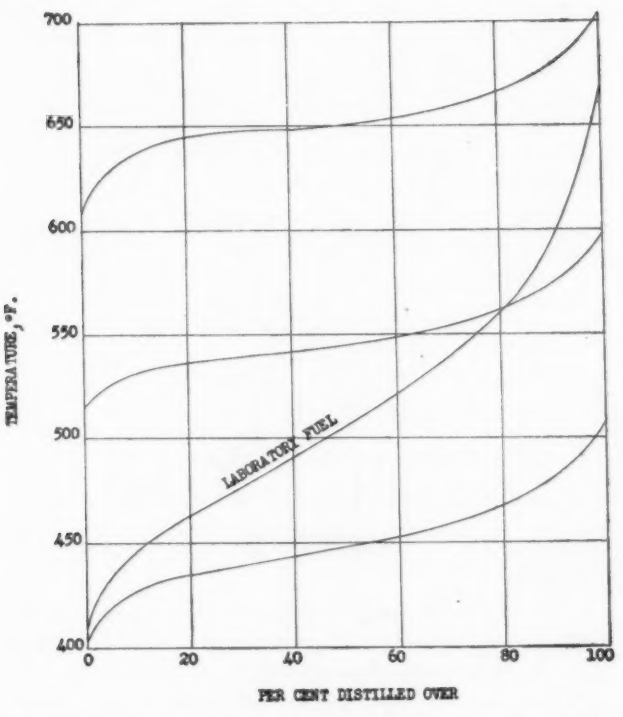
Part I

Results from a Series of Narrow Distillation Range Fuels (Cetane Numbers from 40 to 90)

A DIESEL fuel which has a cetane of 45, at low pour point, suitable viscosity, and a low sulfur content is generally considered superior for most operations of diesel engines in the moderate-speed range. The distillation of most diesel fuels comes within the range from 300 to 750 F. Operators prefer fuels of lower boiling range. Recent demands of the armed forces have caused an increase in production of premium fuels. This has also brought about larger quantities of fuel having a cetane below 40. These lower cetane fuels have presented problems in starting and maintenance which heretofore have not been encountered. Thermally and catalytically cracked fuels for diesel engines have come to the front in recent years. These fuels have the desirable quality of low pour point but the autoignition quality is not comparable to the normal paraffin fuels. Ignition promoters, when added to these cracked fuels, may make a highly desirable product.

It is important that we know which compound in a fuel produces a given effect on the operation of the engine. At present there is such a variety of fuels and the analyses of the fuels are so meager that single factors cannot be studied. Moreover, there is some question as to the practicability of studying each compound which is found in a diesel fuel. In fact, this would be an enormous task because of the number of compounds found in a single fuel. Nevertheless, it is important to know the reactions in the diesel engine to a number of fuels of known origin and composition, especially some of the pure compounds which can be isolated and identified as components of diesel fuels. Hydrocarbon series can be isolated and a few of each series studied to gain first-hand information about their combustion. Until such compounds can be furnished in quantities for practical experiments, we must be satisfied with specialized fuels.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 10, 1946.]



■ Fig. 1 – Distillation curves of special fuels and one sample of laboratory fuel

■ Experimental Fuels

Fuels for these experiments had approximate cetane numbers of 40, 55, 70, and 90. For each cetane number there were three distillation ranges – 400 to 500 F, 500 to 600 F, and 600 to 700 F. Table 1 gives the analyses of the fuels. These fuels give the approximate extremes in distillation and cetane of the usable diesel fuels found on the market. Distillation curves for the fuels are shown in Fig. 1 and are compared with the distillation curve for one sample of our standard laboratory fuel.

The pour points of these fuels are plotted as a function of cetane in Fig. 2. The pour point of these fuels is almost a straight-line function of cetane for each distillation range. The pour points of pure straight-chain paraffin compounds are included in this graph. The number by each circle gives the boiling point of the paraffin compound.

Some technical compounds were used in cetane determination, but this is only a preliminary experiment and must be supplemented with the study of other compounds.

■ Test Engines

Single-cylinder liquid-cooled diesel test engines were used in these experiments. One engine was a precombustion-chamber type with a bore of 3¾ in. and a stroke of 5 in. The compression ratio of this engine was 18 to 1. Windows for observation and pressure pickup outlets in both engine chambers made it possible to take measurements simultaneously in these chambers. The second engine was of the direct-injection type having the same bore and stroke as the precombustion-chamber type. The piston in this engine had a hemispherical crater which served as a combustion chamber while the piston was at the top of the cylinder. With one piston, the engine had a compression ratio of 15 to 1. This piston could be replaced by another having a larger crater to give an engine of compression ratio of 12 to 1. A shrouded intake valve and the piston crater were used to induce turbulence of the air in the cylinder. Diagrammatic sketches of the two types of combustion chambers are shown in Fig. 3.

ENGINE

by E. W. LANDEN
Research Department
Caterpillar Tractor Co.

THE first part of this report on the combustion problems that have been encountered in diesel engines presents the results of experiments conducted on fuels of narrow distillation range and 40 to 90 cetane number.

The second part deals with the diesel flame, revealing data on its intensity, temperature, and spectra.

THE AUTHOR: E. W. LANDEN, who came to Caterpillar Tractor Co. in 1943 from the Armour Research Foundation, has been interested in diesel combustion since 1940. Before that, he instructed physics at the University of Missouri, where he received his Ph.D. in 1938.

Considerable data on the 12 special fuels were taken with the direct-injection engine.

Cetane

Ignition quality of a fuel is of paramount importance because no burning can take place until properly ignited. The fuel must be selected so that it will ignite early in the combustion cycle. Large quantities of fuel may reach the combustion chamber if the fuel is slow in igniting and this leads to an undesirable, rapid combustion in most engines. The cetane number, introduced as a method of rating diesel fuels, gives a measure of ignition quality. Fuels of any cetane number within reason can now be supplied, and cetane improvers may be used to supplement the fuel for even higher values.

The practicality and desirability of high-cetane fuels must be known before engine designers can take full advantage of this type of fuel. In long-term planning, when it becomes necessary to synthesize fuel, this will be more important. Engine design has naturally been centered about a fuel of between 40 to 50 cetane because this has been available and has had the desirable qualities specified by the manufacturer.

Data on straight-run diesel fuels from the oil-producing localities have been reported by Bogen and Wilson.[1] They give the percentage of paraffinic, naphthenic, and aromatic content. In general, the cetane is a straight-line function of the paraffinic content, as shown in Fig. 4. Cloud and Ferenczi[2] have determined cetane for the normal paraffins heptane (55) and dodecane (80), and cetane by definition (100) is known. These values are also shown in Fig. 4. From these values, it appears that the cetane is almost a straight-line function of the number of carbon atoms in this range.

Alkylated benzenes have been studied by Ju and Wood.[3] The normal paraffinic side chain may be considered paraffinic and the benzene may be considered aromatic. On a weight basis, the paraffinic content may be evaluated. Fig. 4 also shows their data on the alkylated benzenes.

The ethers and many ignition promoters, such as alkyl nitrate and peroxides, all have high cetane numbers.

The cetane numbers of a few available compounds were measured by blending with normal cetane. A Caterpillar cetane valve was used for these measurements. Extrapolated values are given in Table 2.

Ju and Wood[3] obtained slightly lower values for benzene and toluene, but they report toluene lower than benzene.

The influence of cetane on specific fuel consumption was measured on the 12 fuels. The direct-injection engine was used with both 15 to 1 and 12 to 1 compression ratios. Fig. 5 shows the average specific fuel consumption as a function of cetane for 95 bmep and engine speed of 1800 rpm. This curve is an average of all trials taken for the two compression ratios. Timing was adjusted so that the beginning of injection varied from 13 to 29 deg of crank angle before top center. The fuel consumption was a function of timing and hence comparison of the minimum fuel consumption for the two compression ratios cannot be made from these data. These data show that these particular engines were more efficient with the lower cetane fuels.

The specific fuel consumption was almost a straight-line function of distillation range. The 600-700 F distillation

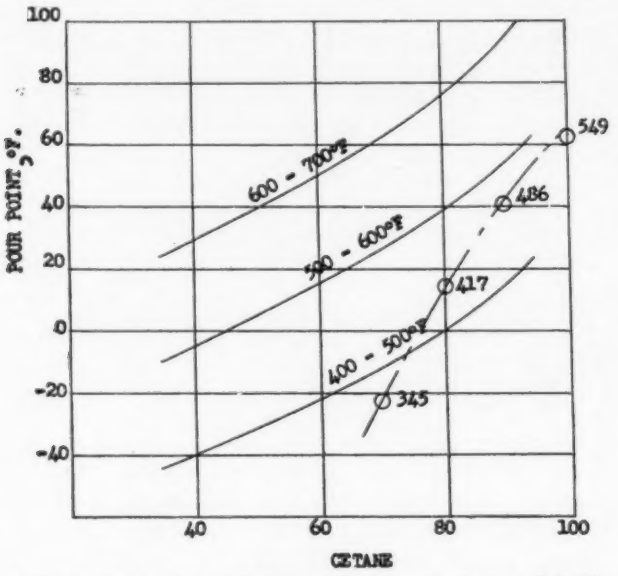
[1] See *Petroleum Refiner*, Vol. 23, July, 1944, pp. 272-282: "Ignition Accelerators for Compression-Ignition Engine Fuels," by J. S. Bogen and G. C. Wilson.

[2] See SAE Journal (Transactions), Vol. 52, June, 1944, pp. 233-237: "Special Cold-Starting Fuels for Diesel Engines," by G. H. Cloud and L. M. Ferenczi.

[3] See *Oil and Gas Journal*, Vol. 39, Jan. 9, 1941, pp. 41, 43-44, 46: "Cetane Number and Chemical Composition," by T. Y. Ju and C. E. Wood.

Table 1 – Characteristics of 12 Fuels Used in These Experiments
(Other factors, omitted from table, are similar)

Cetane	40	42	43	56	56	55	70	72	72	89	92	89
Flash, F	182	255	310	178	255	320	186	255	325	198	255	325
Gravity, API	36.2	29.0	23.1	42.0	35.1	29.6	46.5	40.5	34.3	52.1	48.6	45.2
Pour Point, F (below)	−35	−5	30	−20	15	45	−15	25	55	15	55	100
Aniline Point, F	138	144	142	154	161	170	165	178	185	189	203	226
SU Viscosity at 100 F sec	32.9	41.7	86.5	32.7	39.4	62.0	31.9	37.6	54.0	32.4	36.7	41 at 130
IBP	401	517	606	405	511	601	410	518	618	422	498	606
10%	431	532	633	430	532	639	429	532	634	442	522	639
50%	448	545	650	448	545	652	446	546	648	457	542	654
90%	481	568	679	480	571	681	476	570	673	479	576	690
FBP	516	585	698	525	598	709	505	591	694	502	592	711



■ Fig. 2 – Four points of fuels as a function of cetane and distillation range

range fuels had on the average 4% higher specific fuel consumption than the 400-500 F distillation range.

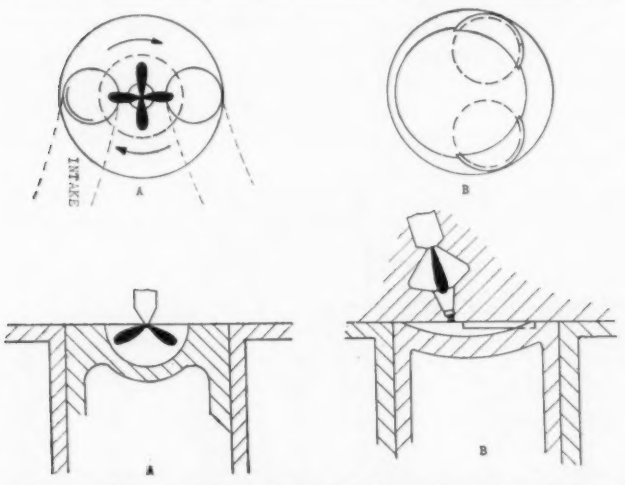
■ Flame Initiation and Duration

Pictures of the ignition process and combustion of injected fuels taken by Rothrock and Waldron[4] show that ignition takes place near the bounding edge of the injected spray. Pictures by Holfelder[5] also show this fact to be true. The luminosity of the burning in the fuel spray means that the entire reaction has reached an advanced stage. The author has taken photographs of a fuel sprayed into a precombustion chamber, and Fig. 6 shows some of these. They show that individual fuel droplets injected into the heated atmosphere ignite and burn with luminous combustion. Sufficient luminosity is produced so that they can be photographed by the light radiated during the burning. The droplets move through the chamber at a high velocity and, therefore, are continually entering a new oxygen supply. As the drops move, they expose the film and each produces a streak of light. These droplets are remote from one another but they show similar reactions. Thus, 50 or 100 reaction zones are visible at one time. In the photographs where large quantities of fuel are injected, the individual burning droplets overshadow each other and we have a single burning mass. In fact, when slightly more fuel is injected into the chamber than that necessary for showing individual droplets, the entire chamber seems to be filled by a sheet of flame. While perceptible pressure could not be noted when the visible burning droplets were in the chamber, a pressure was observed on the card when slightly greater amounts of fuel were injected, so that the chamber was filled with flame.

The temperature of the burning droplets is high, although there is little effect on the average pressure within the chamber. Hence, localized temperatures may become very much higher than the average temperature of the gases in the chamber. This is especially true in the early part of the combustion cycle. The luminosity of the burning is not an indication of the temperature but it does indicate that particles are heated sufficiently to show incandescence. Since the fuel is injected into a heated atmosphere which is still further heated by the burning, the pyrolytic action causes a breakdown of the fuel by cracking, which is probably responsible for the greater part of the carbon. In the cracking reactions, especially when olefins are produced and oxygen is not properly distributed, polymerization reactions also take place. Polymerization products which have slower rates of reaction lead to incomplete combustion.

[4] See NACA Report No. 561 (1935), "Effect of Nozzle Design on Fuel Spray and Flame Formation in High-Speed Compression-Ignition Engine," by A. M. Rothrock and C. D. Waldron.

[5] See NACA Technical Memorandum No. 790 (1936), "Ignition and Flame Development in Case of Diesel Fuel Injection," by O. Holfelder.

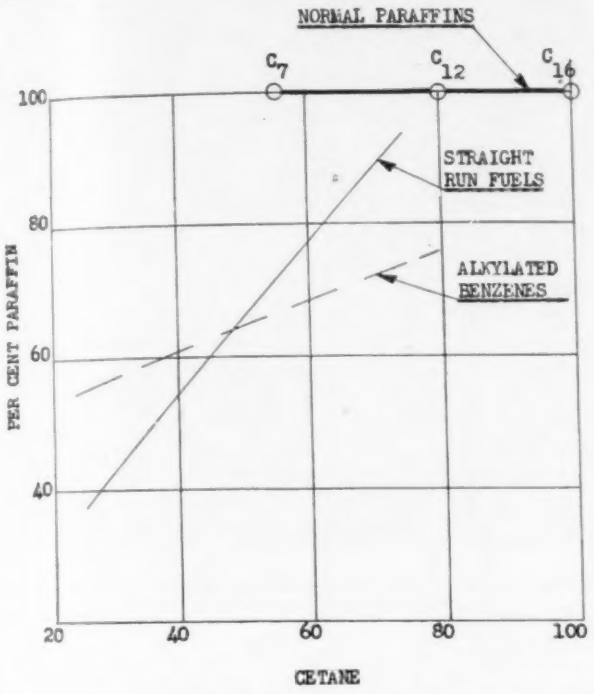


■ Fig. 3 – Diagrammatic sketch of direct-injection engine shown at *A*; that of precombustion-chamber engine shown at *B*

Table 2 – Extrapolated Cetane Numbers of Compounds

Compound	Cetane Number
Cyclohexane	13
Benzene	0
Toluene	−5
Xylene	−10

The combustion flame in a compression-ignition engine is characterized by high luminosity throughout the burning period when fuel is injected into the precombustion chamber of a two-chamber engine. The flame entering the main chamber has a high luminosity at the outset and continues to burn vividly for a time and then the luminosity decreases until it is extinguished. The duration of the flame is determined primarily by the engine speed and the amount of fuel injected. Flame trace photographs from the two chambers show the flame initiation and termination as a function of engine load. At small values of bmep, the observable flame in the prechamber lasts for a longer time than the flame in the main chamber. Loads near the maximum indicate that the flame in the prechamber is extinguished earlier in the cycle than the flame in the main chamber. Fig. 7 shows some flame photographs taken from the two chambers at various loads. Fig. 8 shows graphically the flame duration as a function of bmep for the two chambers. The flame appeared in the main chamber about 0.25 millisec after it appeared in the prechamber at 1800 rpm with a fuel of 48 cetane. The



■ Fig. 4 – Cetane number of fuels as function of paraffinic content

cetane has some effect on the delivery from the one chamber to the other. This time interval is determined by the rate of reaction in the prechamber and the restriction between the chambers. Fig. 9 shows two photographs of simultaneous pressure cards taken from these two chambers. One pressure card is inverted. If sufficient fuel is injected into the engine chambers, combustion may not be completed and smoke will emerge with the exhaust. In fact, the combustible mixture may still be burning when it emerges from the exhaust port.

The direct-injection engine used in this experiment had one window situated so that one fuel spray passed across the axis of the window. The early stages of luminosity could then be observed but this may not be the first luminosity in the combustion chamber. Burning may start in places where observation was impossible. Photographic data on the flame of the direct-injection engine given by Landen and Blanc[6] show the duration of flame as a function of load. In this engine, the flame seems to be divided into two parts: the early stage having a very high luminosity and the later stage having a lower luminosity. Additional data taken with a photoelectric cell verify these results. Fig. 10 shows the flame duration as a function of bmep for the direct-injection engine having a compression ratio of 12 to 1.

Loading the engine sufficiently caused a long, lingering flame in the direct-injection engine chamber. Flame could be observed in the combustion chamber until the exhaust valve opened, at which time the flame came out of the exhaust port. In this case the window became sooty because of the heavy carbon formation during the combustion. Smoke emerging from the exhaust port increased rapidly when the angular value of flame termination neared the opening of the exhaust valve. Fig. 10 also shows the smoke as a function of the bmep. These readings were taken at 1800 rpm, and it was assumed that the smoke quality and particle size remained the same for all loads. The smoke quality did not change visually with change in load.

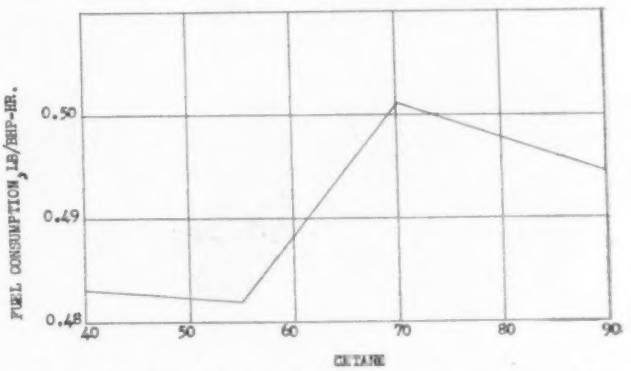
The smokemeter used in these experiments, which has been described by Brown,[7] consisted of an electric lamp placed at one end of a tube 28 in. long and a photoelectric cell at the other end. The photoelectric current was measured by a microammeter. A sampling tube with five ¼-in. holes along the length extended across the exhaust pipe about 3 ft from the exhaust port. The holes faced the engine port. Approximately 6% of the exhaust gases, including the smoke, was drawn through the smokemeter tube by a small blower. In Fig. 10, and the remaining figures, the exhaust smoke will be indicated by the equation

$$n = K \log \frac{I_o}{I}$$

where n is the number of smoke particles in the smokemeter tube, K is a proportionality constant, I_o is the light intensity measured through the clean smokemeter tube, and I is the light intensity as measured with smoke passing through the smokemeter tube. This method of representation was chosen in preference to a per cent method because very high values of smoke density were measured. For low values of smoke the per cent and n are parallel but diverge rapidly above 50%. Smoke color was determined visually, since no provision for measuring the color was made on the smokemeter.

Combustion flame duration of a fuel in an engine cylinder depends on the cetane number. Using the direct-injection engine, it was found that the higher cetane fuels produce a longer visible flame in the combustion chamber. The duration of luminous combustion as a function of cetane number is shown in Fig. 11. A photoelectric cell was used to measure the luminous intensity. Robertson et al[8] have also reported this to be true for a commercial, two-chamber engine.

The paraffinic high-cetane fuels may produce more carbon in early stages of combustion than low-cetane fuels and hence take longer for the carbon to be consumed in



■ Fig. 5 – Specific fuel consumption for various cetane-number fuels – 1800 rpm, 95 bmep, direct injection

[6] See ASME Transactions, Vol. 65, February, 1943, pp. 121-125: "Instrumentation in Study of Diesel Combustion," by E. W. Landen and L. A. Blanc.

[7] See SAE Journal (Transactions), Vol. 48, May, 1941, pp. 188-192: "Requirements of Smokemeter," by K. M. Brown.

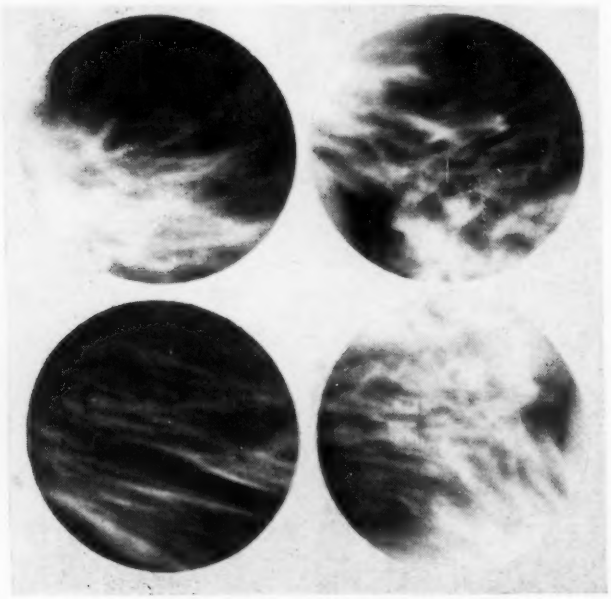
[8] See SAE Journal (Transactions), Vol. 44, March, 1939, pp. 117-124, 140: "Duration of Combustion in Commercial Diesel Engine," by A. F. Robertson, R. A. Rose, and G. C. Wilson.

the cylinder. This phenomenon may explain the longer flame for the higher cetane fuels and also, as shown later, causes more black smoke. It is well known that high-cetane fuels are paraffinic in nature. Since the paraffins are least stable thermally, the high-cetane fuels are most susceptible to the pyrolytic action in the combustion chamber. This cracking process precipitates carbon from the fuel. The particle size and method of agglomeration are not very well known but data will be presented in Part II of this paper to show that the precipitate is very dense. It is the incandescent carbon which largely contributes to the luminosity of the flame. If sufficient quantities are precipitated, the luminous combustion continues until it reaches the exhaust port and a black soot will emerge.

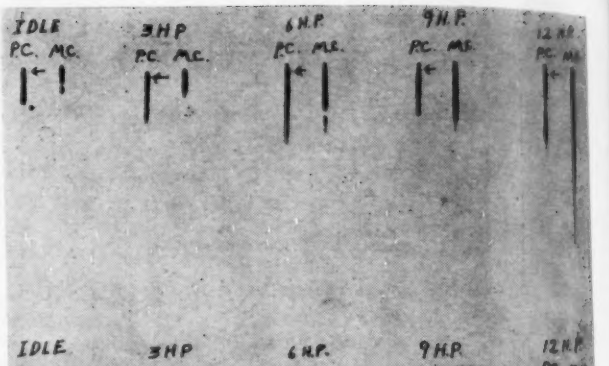
The distillation range influenced the duration of combustion slightly, if at all. At 12 to 1 compression ratio, the higher distillation range fuel indicated a slight increase in flame duration when compared to the low distillation range. At 15 to 1 compression ratio the reverse was found to be true so the experiment is not conclusive. The average value from both compression ratios shows no change in flame duration with distillation range.

■ Smoke Measurements

Exhaust smoke measurements were made from each fuel sample while burning in the 12 to 1 compression ratio engine. The method of measuring exhaust smoke was given in the previous section. Engine operating conditions were maintained as nearly constant as possible at 1200 rpm. The effect of engine timing on the production of smoke was determined with a cracked fuel of 50 cetane. Exhaust smoke, as a function of bmep, is shown in Fig. 12 when the fuel injection setting is changed. Retarded timing causes smoke to appear at smaller values of bmep than does advanced timing. When the timing has advanced to a certain degree, a further advance does not appreciably change the smoke.



■ Fig. 6 – Burning droplets of fuels having sufficient luminosity to photograph produce streaks of light (taken through 1/4-in. window in precombustion chamber)



■ Fig. 7 – Flame trace photographs showing flame duration in precombustion chamber and main chamber

The fuels having a distillation range of 400 to 500 F produced very little smoke at small values of bmep. The 40-cetane fuel produced the least smoke but a smoky condition occurred at about 75 bmep and above. As higher cetane fuels were used, the smoky condition occurred at smaller values of bmep, as shown in Fig. 13. When the fuels having a distillation range of 600 to 700 F were burned, it was found that the 40-cetane fuel was very smoky, especially at small values of bmep. These fuels produced less smoke at a given bmep as the cetane number of the fuel was increased. The intermediate distillation range fuels, 500 to 600 F, were slightly more smoky than the lower distillation range fuels but produced much less smoke than the fuels of high distillation range.

The exhaust smoke from the low distillation range fuels was black, while that from the high distillation range fuels had a bluish cast. Since the lighter, more-volatile fuels have smaller molecules, they are apparently more completely oxidized. Also, fuel deposited on the piston during injection would evaporate at a rate determined by volatility, air turbulence, and the temperature of the piston. Furthermore, the higher cetane fuels, being more paraffinic, tend to be cracked more easily by pyrolytic action during combustion. Thus, greater quantities of carbon are precipitated which are not altogether consumed during combustion. Some of it emerges as black carbon smoke. The higher boiling, less-volatile fuels produced a blue smoke. Condensate of the exhaust shows that an ether extract is composed of an oily portion and a thick, dark, tarry substance. The oily portion is probably fuel which did not burn, while the tarry substance must be either polymerization products or the extreme heavy ends. The filtered water separation from the condensate when evaporated to dryness had an appreciable quantity of black material. All these extractions had an odor characteristic of an exhaust system. The condensate from the 40-cetane low-boiling fuel had only a small quantity of oily substance and no tarry product. A negligible amount of material came from this water separation. The more paraffinic fuels in the higher boiling range crack and oxidize more readily and the resulting products are more completely burned. Wet-

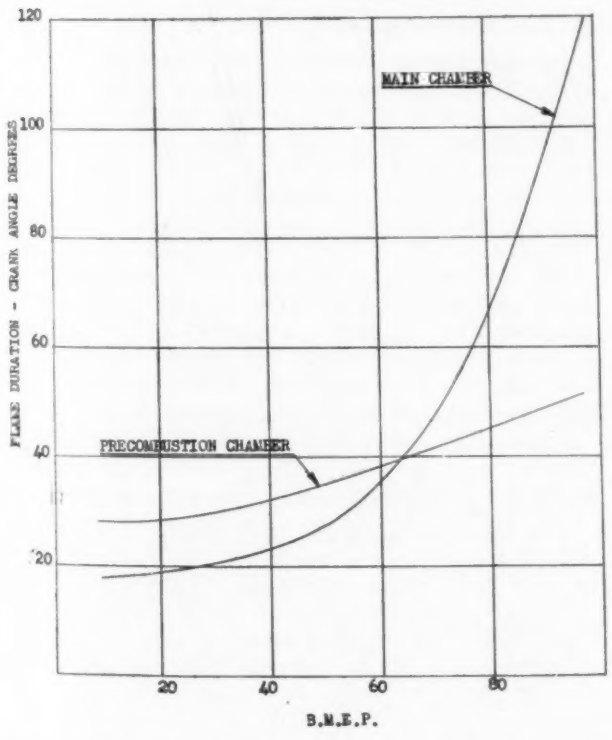
miller and Endsley[9] also show that higher cetane fuels produce smoke at smaller values of fuel injection and hence lower values of bmep. As the fuel quantity decreased, the smoke increased, and in one case misfiring was reported. This reaction is similar to the higher boiling, 40-cetane fuel mentioned above.

Blending the fuel of distillation range 400 to 500 F, shown by *A* in Fig. 14, with the fuel of distillation range 600 to 700 F, shown by *B*, gave intermediate values of smoke. Values in the figure are from 40-cetane fuel, which was found to be least offensive in the low-boiling fuel and most offensive in the high-boiling fuel.

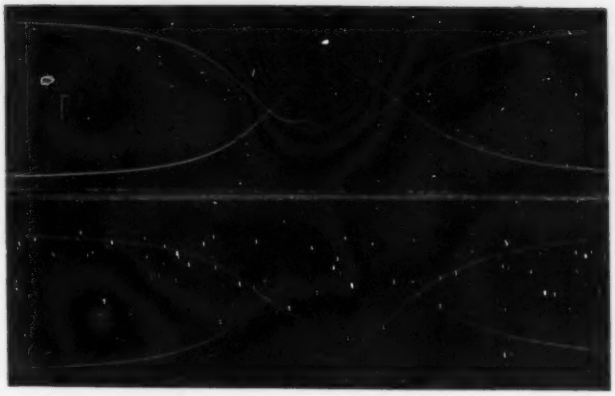
Blends were made of cetane and benzene and cetane and alpha methyl naphthalene to test the effect of adding aromatic compounds to normal paraffin fuel. The cetane did not produce much smoke at small values of bmep but did produce heavy smoke at high bmep. Neither the blend with benzene nor the blend with alpha methyl naphthalene changed the exhaust smoke appreciably in the 12 to 1 compression ratio engine. Both blends were of about 50 cetane compared to 100 for cetane alone. All these compounds have low-boiling characteristics.

One experiment was performed in which exhaust smoke was measured on the low-boiling 40-cetane fuel with cetane improver added. The improver did not change the smoke at small bmep but made smoke appear at a smaller bmep than did the fuel alone. Fig. 15 shows these values. Smoke measurements were made with 2% and 20% improver added to the fuel. A blend of the high-boiling fuel and 20% improver gave less smoke than the fuel alone. These values are also shown in Fig. 15. It is interesting to note

[9] See SAE Journal (Transactions), Vol. 50, December, 1942, pp. 509-520: "Effect of Diesel Fuel on Exhaust Smoke and Odor," by R. S. Wetmiller and L. E. Endsley, Jr.



■ Fig. 8 – Flame duration in precombustion-chamber engine – 1800 rpm



■ Fig. 9 – Pressure cards from precombustion chamber, base line at bottom, and main chamber, base line at top – upper photo: 1800 rpm, 60 bmep; lower photo: 900 rpm, 60 bmep

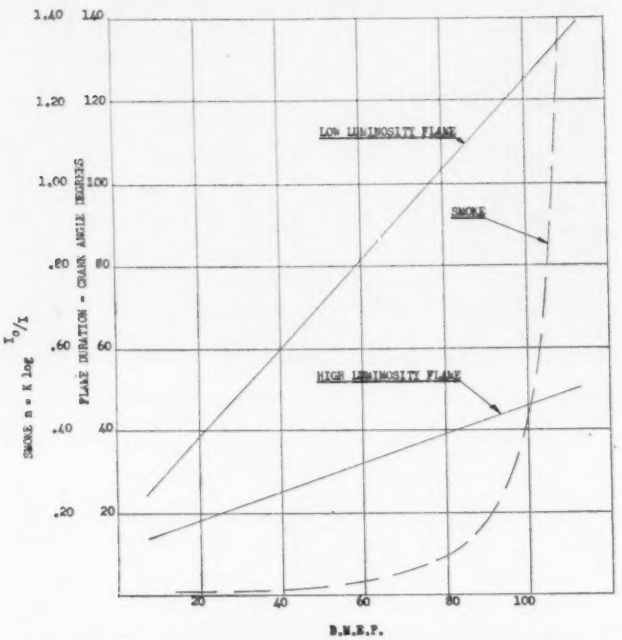
that the cetane of the blended fuels, that is, 70 cetane, with 20% cetane improver added, gave smoke readings similar to the 70-cetane straight-run fuels. However, the 600-700 F distillation range fuel when diluted with the lighter 400-500 F fuel gave a reduced smoke reading. The initial boiling point of the improver was 250 F and the 50% point was 380 F. Therefore, diluting the 600-700 F fuel with the low-boiling cetane improver would produce less smoke, regardless of its cetane quality. On the other hand, there was a definite increase in smoke with the lighter fuel when the improver was added. This must be attributed to a change in the cetane.

■ Ignition Delay, Rate of Pressure Rise, and Peak Cylinder Pressure

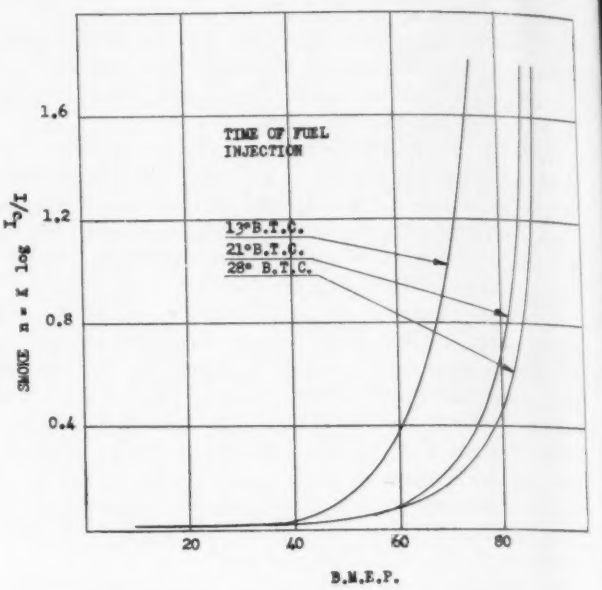
Ignition delay was measured by the time interval from the opening of the fuel valve needle until the combustion pressure rise was observed on the pressure card as presented by an oscillograph. To clarify the method of measurement, Fig. 16 shows a pressure card with various markings on it. From left to right the angle marker is shown at 30 deg before top center. Next, an impulse on the card shows when the fuel valve opens and another one shows when it closes. Then a balanced diaphragm pressure indicator caused an impulse as the pressure in the engine cylinder increased and again when it fell below the gas pressure on the opposite side of the diaphragm.

The angle marker was caused by the illumination of a strobotron falling on a photoelectric cell. Light from the strobotron also fell on the flywheel, so the exact crank angle could be read directly. This angle marker could be adjusted to any desired crank angle value and could be made to coincide with any of the other marks on the card. Then, the procedure was to adjust the angle marker to coincide first with the fuel injection mark and then with the start of pressure rise. The flywheel readings from these settings gave ignition delay in terms of crank angle. Engine speed of 1800 rpm was then used to find the delay in milliseconds. The bmep was approximately 95 for all trials. Fig. 17 shows the ignition delay when injection started at 23 deg BTC. Cylinder pressure at start of fuel injection was 265 psi for 15 to 1 compression ratio and 235 psi for the 12 to 1 compression ratio.

The higher distillation range fuels had an average ignition delay about 4% greater than the lower distillation range fuels.



■ Fig. 10 – Flame duration and smoke – 1800 rpm, direct injection



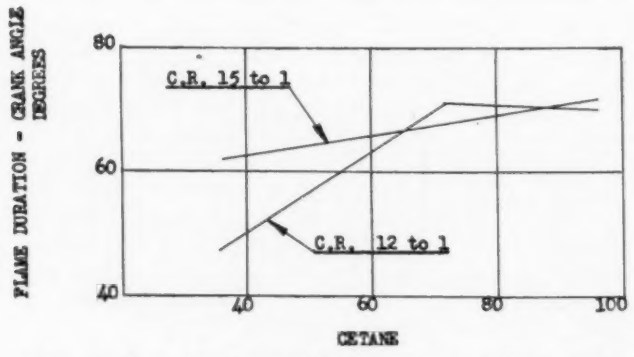
■ Fig. 12 – Effect of timing on exhaust smoke – 1200 rpm, direct injection

The pressure card shown in the figure was produced by a quartz crystal, amplifier, oscillograph, and a drum camera. A second amplifier and oscillograph amplified the photo current obtained from the luminosity of the combustion. The two images were superimposed by placing a 50% reflecting mirror at a 45-deg angle in front of the camera lens. Thus, the light from one cathode-ray tube passed through the mirror and the light from the second cathode-ray tube was reflected into the lens.

Rate of pressure rise was measured by a resistance-capacitance differentiating circuit between the amplifier and oscillograph. Table 3 gives the rate of pressure rise for engine speed of 1800 rpm and 95 bmep.

The data show that rate of pressure rise is lower for higher cetane fuels and is higher in the direct-injection engine as the timing is advanced. The higher compression ratio engine had a slightly higher rate of pressure rise for a given fuel and a given timing.

[10] See *Chemical Reviews*, Vol. 22, February, 1938, pp. 61-85 + (disc.) 85-87: "Combustion Process in Diesel Engine," by G. D. Boerlage and J. J. Broeze.



■ Fig. 11 – Flame duration as affected by cetane number in direct-injection engine

Peak cylinder pressure became higher: as the cetane of the fuel was lower, as the timing was advanced in the direct-injection engine, as the compression ratio was increased.

The volatility of these fuels seemed to have a very slight effect either on the rate of pressure rise or the peak pressure as measured in these experiments.

■ Acknowledgment

Th author wishes to acknowledge the helpful criticism of Dr. L. A. Blanc in the final preparation of this paper. He also wishes to thank C. H. Paul and R. L. Compton who assisted him in taking some of the data.

Part II

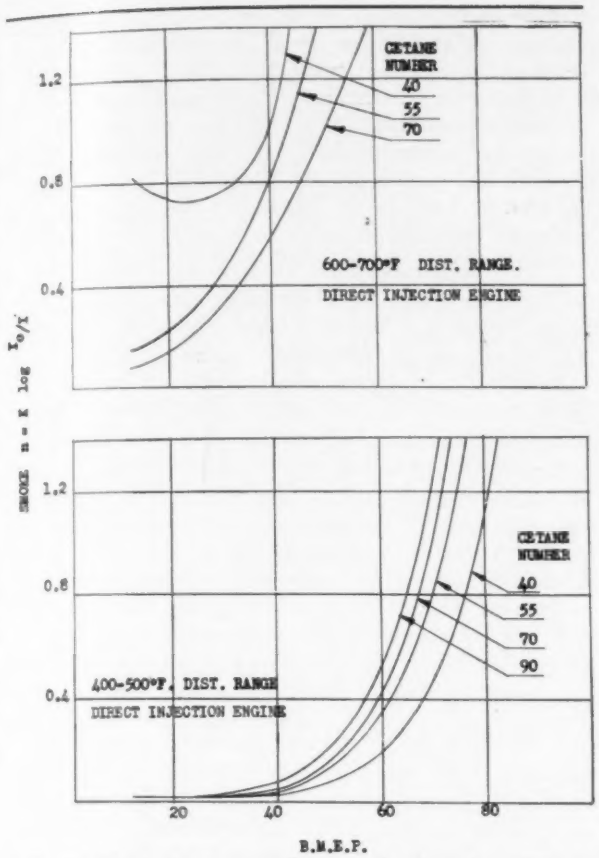
Intensity, Temperature, and Spectra of Diesel Flame

The first part of this paper gave some quantitative results on fuels having specific characteristics. The second part will be more qualitative in nature because the work is still in progress and only a few quantitative results have been obtained. Elaboration and repetition of some points already reported in the literature may be helpful in gaining a clearer picture of the combustion in the diesel engine.

■ Flame Intensity

A bright luminosity is associated with the combustion in the diesel engine and is prevalent throughout most of the burning phase of the combustion. The initial and final oxidation reactions may not have luminosity associated with them. Flameless combustion has been demonstrated in the diesel engine by Boerlage and Broeze,[10] but in current engines under normal operating conditions the flame is very intense.

Liquid fuel entering the combustion chamber initially fills only a fraction of the total volume. The mixing of fuel and air in the combustion chamber must then be

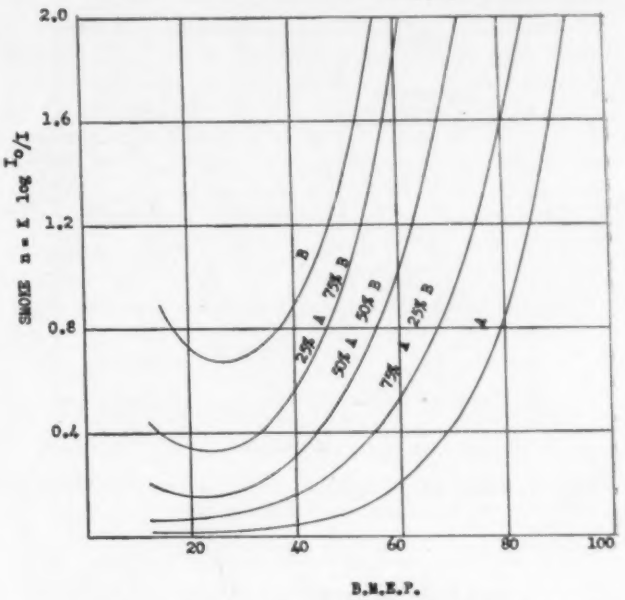


■ Fig. 13 – Effect of cetane number on exhaust smoke – 1200 rpm

handled carefully to utilize both components efficiently. Due to the fact that initially the fuel is surrounded by a limited quantity of air, one would expect some reaction of the fuel vapor other than the normal oxidation reactions. Initial oxidation reactions of the fuel, sometimes called preflame reactions, are very important and the rate at which these reactions proceed is related to the cetane number of the fuel. Other factors influencing this reaction rate are: air temperature, pressure, density, turbulence, and the atomization of the fuel. Localized regions of high temperature appear when the heat generated by the preflame reaction is greater than that lost by the fuel nucleus to the surroundings. The energy liberated by these reactions produces local temperatures which may become 1000 to 2000 F higher than the temperature of the compressed gases.

Fuel injected into the heated air of the combustion chamber immediately starts to evaporate as the first physical reaction. Then the heated fuel vapor and oxygen start reacting at the fuel-air boundaries. As soon as the reactions have proceeded sufficiently and the local temperature is high enough the fuel in the vapor phase begins to crack and precipitate materials with a high percentage of carbon which, by virtue of the localized conditions, becomes heated to incandescence. This incandescent carbon largely contributes to the luminosity of the flame. The precipitated carbon is either consumed during the combustion cycle or emerges from the exhaust port in the form of soot, along with the other burned and unburned products of the fuel.

[11] See *Industrial and Engineering Chemistry*, Vol. 24, February, 1932, pp. 219-222: "Reaction-Velocity Constants of Oil Cracking," by J. C. Geniesse and R. Reuter.



■ Fig. 14 – Smoke produced by blends of 40-cetane fuels – 1200 rpm, direct injection
A – 400-500 F boiling range
B – 600-700 F boiling range

Gas temperatures reached in the combustion chamber during compression vary principally with the compression ratio of the engine and the initial air drawn into the cylinder. These temperatures are in the neighborhood of 1100 F (593 C) in most moderate speed engines. At this temperature the vapor phase cracking reactions are not too rapid for gas oils. Geniesse and Reuter[11] have actually measured the reaction rates for a gas oil and they find that at 1100 F it takes 1.04 sec to convert 10% of the fuel, and at 800 F it takes 1040 sec to convert the same quantity. By extrapolation, from their data, at 1500 F the same conversion takes place in 0.001 sec. Compression pressure effects have been disregarded in the above assumptions. Hence, cracking of the fuel goes on rapidly at elevated temperatures. In the diesel engine these reactions are enhanced by the proximity of fuel vapors to the burning flame in contrast to conditions in the otto-cycle engine where heat transfer must precede a rapidly progressing, well-defined flame front.

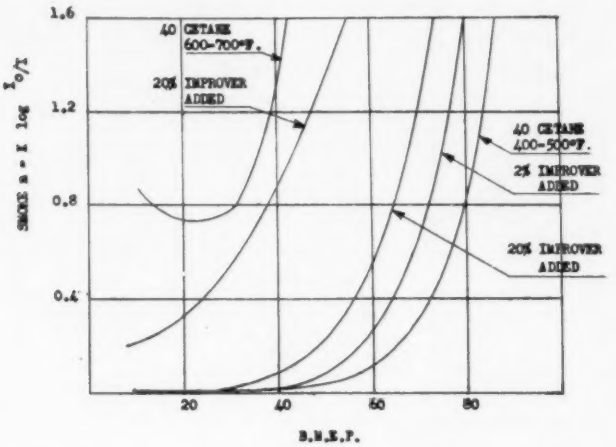
The flame in a high- or medium-speed diesel engine is not uniform in intensity at all times. The depth from which the radiation comes is not too great because of the density of the charge in the chamber.

■ Flame Temperature

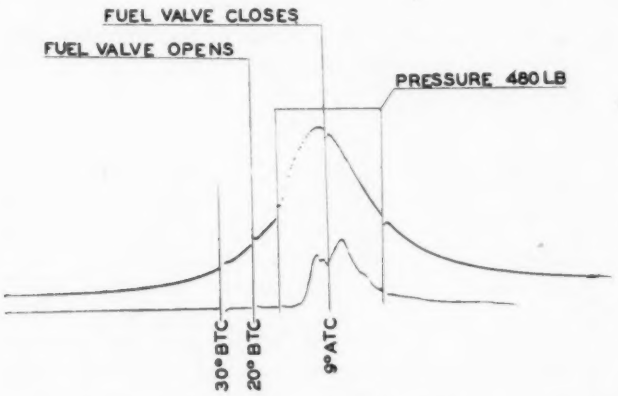
Any temperature measuring device which is inserted into the combustion chamber of an engine is not very satisfac-

Table 3 – Rate of Pressure Rise
(Psi per sec)

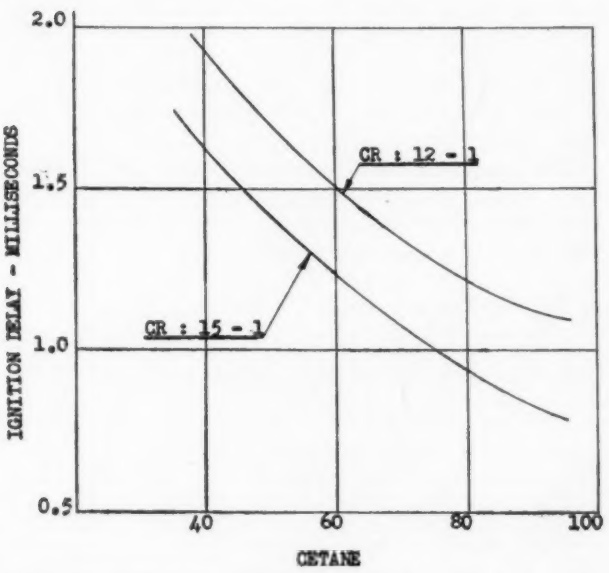
Compression Ratio	Injection Start, deg BTC	Cetane 40	50	70	90
12 to 1	19	1.17x10⁶	0.79x10⁶	0.62x10⁶	0.47x10⁶
12 to 1	29	2.0	1.42	1.07	0.77
15 to 1	19	1.18	0.81	0.68	0.56
15 to 1	29	2.24	1.52	1.18	0.98



■ Fig. 15 – Smoke as produced when cetane improver added to fuel – 1200 rpm, direct injection



■ Fig. 16 – Pressure card from direct-injection engine showing marks used for indicating angles, injection, and pressure – flame intensity shown below



■ Fig. 17 – Ignition delay as function of cetane number for one position of timing – 1800 rpm, 95 bmep, direct injection

tory for measuring the gas temperature. The temperature changes too rapidly for the thermoelement to follow because of the low heat transfer coefficient and the specific heat capacity of the element. Methods which have been used satisfactorily include the spectral line reversal and measurement of radiant energy. These methods are considered practical for temperature measurement on an internal-combustion engine. A number of experimenters have measured the flame temperature in the otto cycle by the sodium line reversal method.[12, 13, 14]

In the otto engine this is satisfactory because the chemiluminescence of combustion is small compared to the intensity of the sodium excitation. However, in the diesel engine the flame intensity is so great that it interferes with the measurement of the intensity of the sodium. Therefore, other means must be used for the diesel flame.

A method of determining spectroscopically the temperature of the diesel flame has been described by Erichsen,[15] who uses Plank's law of radiation. This method gives an average value of temperature for a number of cycles. Schroder[16] has used a photographic method to obtain instantaneous values of temperature during one combustion cycle. While his method is complicated, mechanically and photographically it has virtue from the standpoint of making corrections due to window sootiness. Uyehara et al have recently reported[17] a photoelectric method to measure the radiation at two wavelengths simultaneously throughout the combustion cycle. This method has a number of advantages when compared to the other methods. However, they all give temperature values which are within a range of a few hundred degrees.

It is not surprising that experimenters obtain slightly different temperature values in the diesel flame because each uses a different engine and has different techniques. In burners and the otto engine where the mixture can carefully be controlled, variations in the temperature would be expected depending upon the operating conditions. In the diesel engine the various phases of combustion are not well defined and initially the combustion starts at loci in which there is a comparatively low air-fuel ratio. The flame mixture consists of decomposing hydrocarbons, fuel in various stages of vaporization, carbon precipitate, and decomposition products. The incandescent carbon is the primary emitter of radiant energy.

We have been experimenting with a photoelectric pyrometer for several years. The final model is portable and has sufficient sensitivity for observation. A view of the wavelength selector and preamplifiers is shown in Fig. 18. Associated apparatus and optical system are shown in Fig. 19. Exact values of temperature have not been de-

[12] See SAE Journal (Transactions), Vol. 36, April, 1935, pp. 125-135 + (disc.) 135-136: "Flame Temperatures Vary with Knock and Combustion-Chamber Position," by G. M. Rassweiler and L. Withrow.

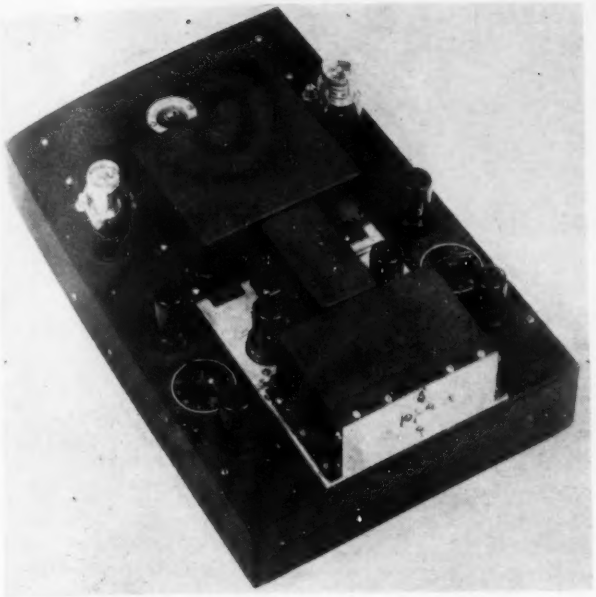
[13] See University of Illinois Engineering Experiment Station, Bulletin No. 262 (1933), "Flame Temperatures in Internal-Combustion Engine Measured by Spectral Line Reversal," by A. E. Hershey and R. F. Paton.

[14] See NACA Technical Note No. 559 (1936), "Combustion-Engine Temperature by Sodium Line Reversal Method," by M. J. Brevoort.

[15] See *Forschungsheft 377*, Vol. 7, March/April, 1936, pp. 1-31: "Spektraluntersuchung des Verbrennungsvorganges," by G. Beck and C. Erichsen.

[16] See *Motortechnische Zeitschrift*, December, 1939, pp. 181-190; January, 1940, pp. 7-15: "Photothermometrische Untersuchung an Dieselmaschinenflammen," by W. Schroder.

[17] See ASME Transactions, Vol. 68, January, 1946, pp. 17-28 + (disc.) 28-30: "Flame-Temperature Measurements in Internal-Combustion Engines," by O. A. Uyehara, P. S. Myers, K. M. Watson, and L. A. Wilson.



■ Fig. 18 – Optical pyrometer wavelength selector and amplifiers – view showing entrance slit and shield boxes for optical system

termined to date but the apparatus has been tried. Our first exposures are reproduced in Fig. 20. From these exposures, necessary adjustments will be made for future pictures and temperature determination. Physical data on the instrument proper will be reserved for a later paper. The continuous radiant energy from the precombustion chamber or other source passes through lenses L_1 and L_2 and is thereby focused on the slit of the instrument. The diaphragm D is used to control the intensity of the spot on the slit. Within the instrument are a lens, mirror, and prism mounted in a Littrow mounting. Two wavelength bands are selected from the spectrum by small oblong mirrors which reflect the energy to the photocells. Each photocell has its individual amplifier and cathode follower which transmit the pulse to the oscilloscope input. Here it is further amplified and thrown on the cathode-ray screen. The two oscilloscopes are placed at right angles to each other and at such distance that the beam will be focused on the film in the camera. A partially coated mirror permits a fraction of the light from the cathode tube in front of the camera to enter the lens. The light from the other cathode-ray tube is partially reflected into the lens of the camera. Both cathode-ray beams can then be photographed simultaneously and, by having only a vertical deflection, as the drum camera rotates it will make the time axis of the two traces coincide. An angle marker on the crankshaft causes an impulse in the trace which shows up on the film.

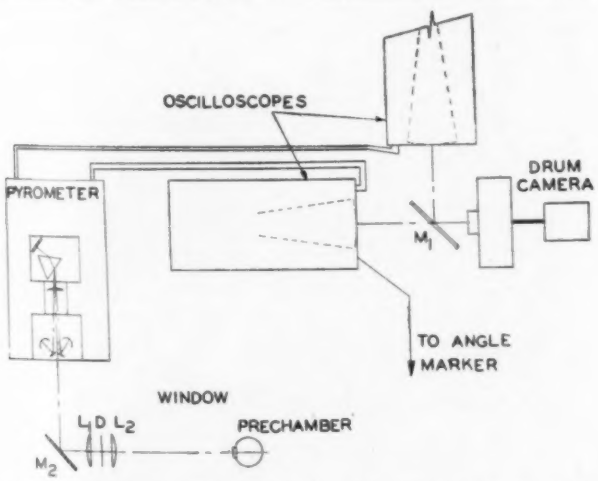
[18] See *Industrial and Engineering Chemistry*, Vol. 18, May, 1926, pp. 528-531: "Ultraviolet Spectroscopy of Flames of Motor Fuels," by G. L. Clark and W. C. Thee.

[19] See *Industrial and Engineering Chemistry*, Vol. 24, May, 1932, pp. 528-538: "Emission Spectra of Engine Flames," by G. M. Rassweiler and L. Withrow.

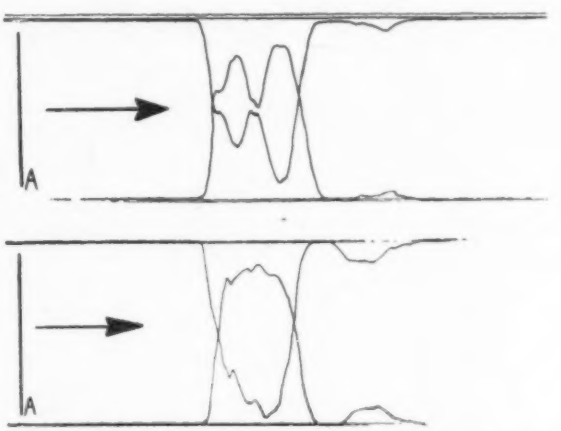
■ Emission Spectroscopy

Emission spectra of the internal-combustion engine have been photographed by a number of observers. Spectra taken from the otto engine by Clark et al[18] and Rassweiler and Withrow[19] show band spectra due to CC, CH, and OH. Spectra from the diesel engine taken by Erichsen[18] show that the diesel flame consists primarily of continuous radiation. The OH band system with band heads at 3064 and 2811 appears in emission spectra. This emission band spectrum is evident in most high-temperature combustion reactions of hydrocarbons. Steam alone, heated to 1200 C or above, shows the OH spectrum in absorption. The CH band spectra are either absent in the diesel engine or do not have sufficient intensity to be observed. The CC bands are not very evident; however, Erichsen reports broad bands in his diesel emission spectra. Chemiluminescense and radiation from incandescent particles are the primary sources of radiation, of which the latter is probably the most important. In the emission spectra taken by the author the only band spectrum observed was due to OH.

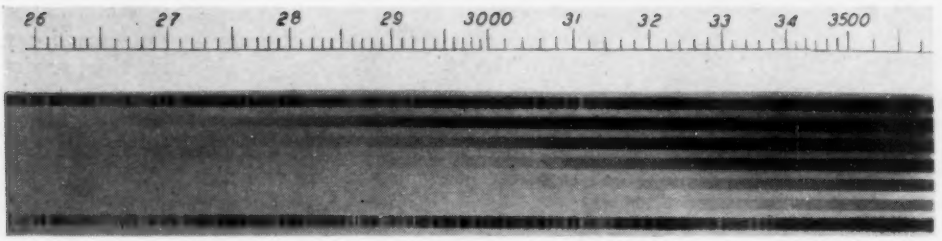
Emission spectra taken with a crystal quartz spectrograph having a focal length of about 5 ft show the OH band heads at 3064 and 2811 superimposed on a background of continuous radiation. Fig. 21 shows spectra, from a precombustion chamber, which have the following exposure times in min: 64, 16, 8, 4, and 1. Continuous



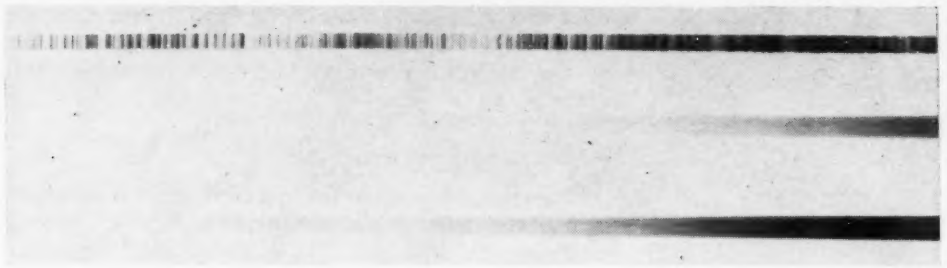
■ Fig. 19 – Schematic view of photocell pyrometer, oscilloscopes, camera, lenses, and mirrors used in determining diesel-engine flame intensity for two wavelengths



■ Fig. 20 – Examples of exposures from two-color photoelectric pyrometer – reading from top base line downward gives intensity of red radiation, reading from bottom base line upward gives intensity of green radiation – A is point at which angle marks appear on oscillogram (retouched)



■ Fig. 21 – Ultraviolet emission spectra from precombustion-chamber flame showing OH emission bands plus continuous radiation



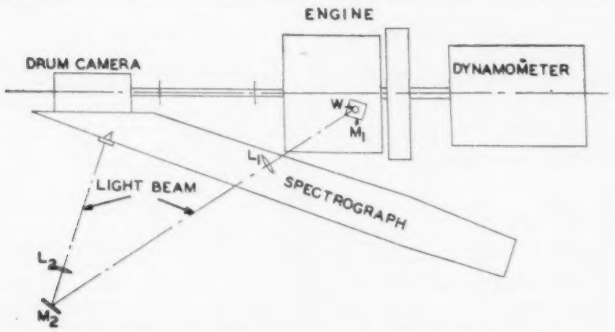
■ Fig. 22 – Uutraviolet emission spectra from direct-injection engine – OH emission band head at 3064 A superimposed on continuous background

radiation in the vicinity of the band head at 3064 is quite high in comparison to the band head itself. A spectrogram of 1 hr exposure taken from a direct injection engine operating at 3 hp is shown in Fig. 22. The intensity of the OH band spectrum in this figure is much greater than the continuous background radiation. This may be the result of the difference in the combustion-chamber design. Either the dispersed fuel mixed with the combustion products produces different emissivity coefficients in the two engines or the actual combustion temperatures differ somewhat to produce this effect.

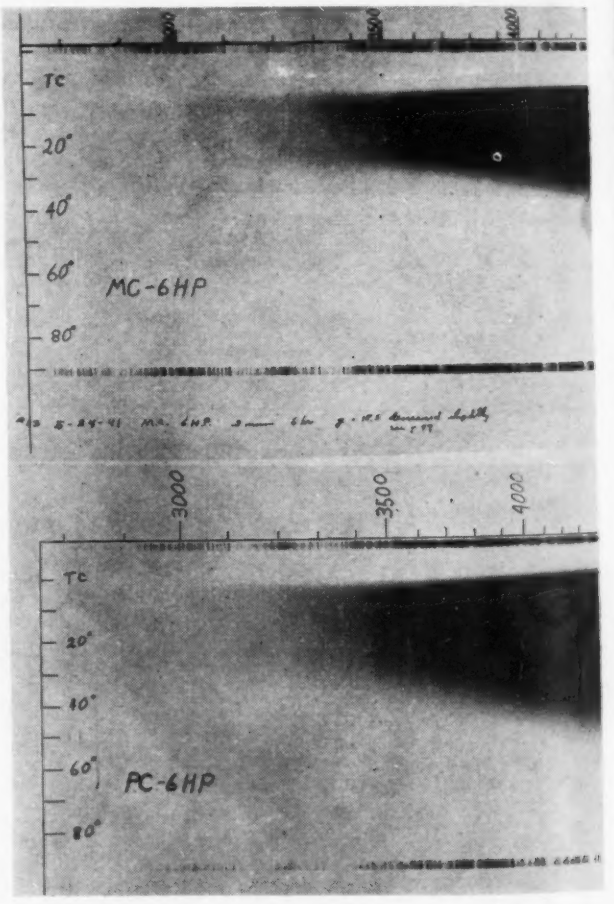
It should be pointed out that the windows for most of these experiments remained quite clean. They were designed so that the combustion pressure wave produced a washing effect to scrub the carbon from the quartz window. The quartz window operated at a higher temperature than the water jacket and, therefore, the carbon did not condense as it would on a colder surface. This window technique has already been described.[20] At heavy loads and high-cetane fuels a film of soot appears on the quartz window when used in a direct-injection engine and main chamber of the precombustion-chamber type of engine.

Some spectra were taken with a drum camera replacing the plate holder and the drum spindle attached to the crankshaft of the engine. The radiant energy was reflected and focused properly so that the spectrograph slit was at some distance from the source. The method is illustrated in Fig. 23. Radiant energy from the window W is reflected by mirror M_1 through lens L_1, which renders the beam parallel. Mirror M_2 reflects the beam through lens L_2, which focuses it on the slit of the spectrograph. The spectrum appears in focus at the surface of the drum camera. Curvature of the spectrum causes some

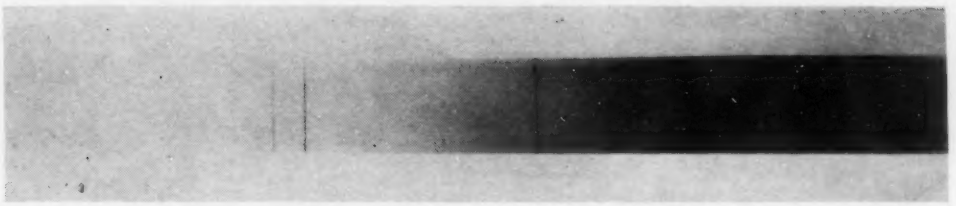
[20] See *Glass Industry*, Vol. 24, January, 1943, pp. 21-23, 33: "Glass Windows for Iron Engines," by E. W. Landen.



■ Fig. 23 – Method of taking spectra of diesel flame with drum camera connected to crankshaft of engine



■ Fig. 24 – Emission spectra taken with drum camera – upper exposure from main chamber and lower exposure from precombustion chamber



■ Fig. 25 – Emission spectrum from precombustion chamber showing lead lines – 1% lead as lead stearate added to fuel

portion of it to be out of focus, since the drum was cylindrical in shape. A direct coupling from the crankshaft to the shaft of the drum ensured exact synchronism at all times. Two spectrograms taken by this drum camera are shown in Fig. 24. One spectrogram from the precombustion chamber is shown below and another from the main chamber is shown above. The duration of radiation from the precombustion chamber lasts longer in terms of crank angle degrees than does the one from the main chamber. At this engine load, these data agree with flame duration measurements reported in Part I. OH band spectra are evident in both spectrograms. The fact that these two spectrograms were taken with different fuels does not invalidate a comparison but brings out another important point. Traces of copper from the refining technique were left in one fuel and the two dark lines in the upper spectrogram are two persistent lines of Cu I, 3247 and 3274 A.

Another spectrogram, Fig. 25, shows three emission lines of lead as produced when 1% of lead is added to the diesel fuel. It was added in the form of lead stearate and the three persistent lead emission lines which appear in this spectogram have wavelengths of 4058, 3683, and 3640 A, respectively. As an illustration of the intensity of radiation and technique, this spectrogram is only a 4-min exposure with the engine idling at 1200 rpm. The amount of lead added caused the engine to knock loudly. An oxide coating formed on the window, which normally remained clean under these operating conditions.

■ Absorption Spectra

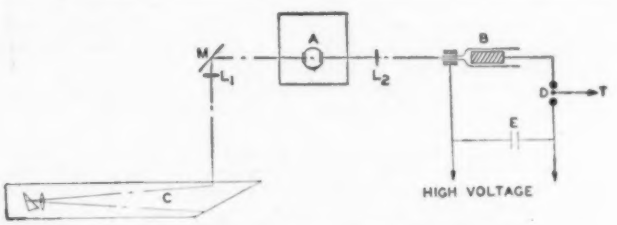
The highly luminous flame of the diesel engine indicates chemiluminescence and radiation from incandescent particles. It is believed that the luminosity comes primarily from the heated carbon precipitated by the explosion. Measurement indicates that there is a heavy precipitate of carbon at the outset of luminous combustion.

A 3¾-in. bore, 5-in. stroke, water-cooled precombustion-chamber engine was used for this experiment. The precombustion chamber was equipped with two windows diagonally opposite, thus forming an absorption cell suitable for taking absorption spectra. Since it is known that emission spectra show continuous radiation plus the band spectra due to OH, these were expected to show up in absorption spectra. This would throw some light on how much carbon is precipitated, and how soon, after the rapid combustion reactions have started.

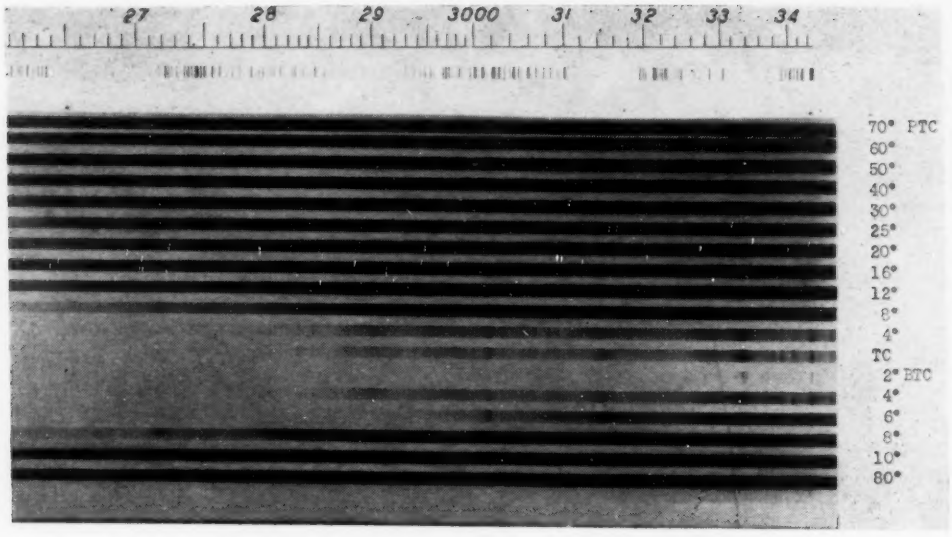
The experimental arrangement for making absorption spectra is shown in Fig. 26. In this figure, *A* represents the precombustion chamber, *B* the essential parts of the capillary discharge tube, and *C* the spectrograph. At *D* is a spark gap which was electrically broken down by a small electrode situated between the main gap electrodes. The small electrode was actuated by a spark coil and commutator connected to the crankshaft by a gear reducer. Discharge of condenser *E* through the capillary occurred only during the firing cycle when combustion products were present. One hundred to three hundred discharges through the capillary were sufficient to give a satisfactory exposure, depending on the photographic density desired. Crank angle position of the discharge could be adjusted manually while the engine continued to operate.

The spectrograph used for taking these absorption spectra was a crystal quartz Littrow type of instrument having an aperature of f:25 and focal length of about 5 ft. The optical path shown in Fig. 26 consists of lens L_1, the absorption chamber *A*, mirror *M*, and lens L_2.

One series of absorption spectra is shown in Fig. 27. The first spectrum taken 80 deg BTC indicates the full intensity of the source passing through the precombustion chamber. The second spectrum taken 10 deg BTC shows very little difference in transparency. However, those taken at 8 and 6 deg BTC show greater absorption, which is due to the amount of fuel entering the chamber. At 4 deg BTC the combustion has started and the fuel is somewhat dispersed and more transparent. Start of combustion is evident by OH absorption bands at 3064 and 2800 A, which are both visible on the plate. Even though the combustion has started and the temperature has reached 1200 C at this crank angle the overall absorption has not changed considerably. Two crank angle degrees later (2 deg BTC) the absorption is almost complete, indicating that additional material has become evident in the absorption chamber. The absorption and scattering of the light by the fuel itself have started to decrease from 6 deg BTC to 4 deg BTC. Knowing that only a small amount of fuel was injected for these exposures, had combustion not occurred, the chamber would again become transparent. The other material which scatters and absorbs the radiation much more strongly than the fuel is the result of the combustion and cracking reactions. The intermediate compounds of initial combustion would probably show an absorption which would be only slightly different from the fuel itself. The simple combustion products would have less complex fuel molecules, CO, CO_2, H_2O, and some other minor constituents, all of which would be transparent except the H_2O. Polymerized products may show strong absorption. The aromatic components of the fuel would have strong absorption but no



■ Fig. 26 – Method of taking absorption spectra through precombustion chamber



■ Fig. 27 – Absorption spectra through precombustion chamber – crank angle at which each spectrum was taken shown at right, wavelengths shown above spectra

complete absorption picture was taken during the interval of fuel evaporation.

The dense carbon precipitate at 2 deg BTC has cleared somewhat from the chamber at top dead center and continues to pass out of the chamber. Pressure measurements in the precombustion chamber reveal that the explosion pressure leaves the compression pressure curve at one crank angle and about 2 deg later had reached peak pressure. This particular chamber had a small orifice which restricted the flow from the precombustion chamber to the main cylinder. At 8 deg ATC the chamber was almost transparent again so far as the carbon precipitate was concerned. The burning lasted until 20 deg ATC as indicated by the OH bands. This was verified by flame intensity measurements. In this spectral region there was no perceptible photographic exposure from the flame itself.

The spectrogram shown in Fig. 27 was taken two years ago, but additional spectrograms are planned for future work in which quantitative measurements on absorption and scattering coefficients will be obtained. Additional absorption spectra would add information about the carbon precipitate. It would be interesting to know whether for longer injection periods additional carbon is precipitated from the fuel throughout the injection period. Air movement complicates the interpretation to some extent. Cyclic variations of combustion are also encountered.

DISCUSSION

Discusses Part I Of Author's Paper

—G. C. WILSON

Universal Oil Products Co.

CATALYTIC Cracking – Some cracked distillate will come from catalytic cracking units. Erroneously, it is a common belief that catalytic cracking will reduce greatly the ignition properties of a diesel fuel, because of the high-octane gasoline produced by these units.

However, from the few cetane-number determinations given in Table A, it can be seen that high-cetane fuel can be produced from high-cetane stock. In general, the damage to ignition properties of a straightrun gas oil is appreciably less by catalytic cracking than by thermal cracking. The samples for the data of Table A were obtained while the plants were still operating to produce aviation gasoline. One reason for expecting catalytic cracking to reduce the ignition qualities of a diesel fuel is that the process tends to produce olefinic stock. However, the operating conditions of a catalytic cracking unit may be controlled over comparatively wide limits, so that the properties of the products can be altered within limits.

Ignition Qualities as Related to Hydrocarbon Structure – Griep and Goddin[a] have emphasized the need for fundamental research that will establish relationships between hydrocarbon structure and ignition qualities. In order to review the known facts about the general grouping of hydrocarbons, they may be listed as follows:

1. The paraffins have the highest ignition qualities. The normal (straight-chain) paraffins are better than the iso (branched) paraffins. The ignition quality of straight-chain hydrocarbons improves as the number of carbon atoms in the chain increases. At least, this is true up to 16 carbon atoms, which is cetane ($C_{16}H_{34}$). Few, if any, ratings have been made for longer chains because of the difficulties encountered in their synthesis.

2. The olefinic (double bond) compounds have medium ignition properties, which in some instances may be quite high.[b] For example, normal cetane ($C_{16}H_{32}$) has a cetane number of 88.[3c]

3. The aromatic compounds, which may be considered as derivatives of benzene (C_6H_6), are known to have very poor ignition qualities.

4. The naphthenic compounds are believed to rate about the same in ignition properties as the aromatics. Boerlage and Broeze[b] considered them to be poorer in this respect than the aromatics, while Ju and Wood,[3] judging from the fact that the naphthenic compounds are less stable thermally than the aromatics, would expect them to rate higher than the aromatics. Both of these two classes of compounds have an advantage in the fact that, in general, they have low pour points.

Petrov and other Russian scientists[c] have gone beyond us in the matter of synthesis and testing of pure hydrocarbons in the diesel fuel range. While their ignition-quality determinations have been made in a bomb and reported in terms of cetene number, there is agreement with our knowledge of the more general groupings. Their results show that the isoparaffins with T-like structure containing

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[a] Numbered footnotes refer to Landen references.

[a] "Significance of Cetane Number in Fuels," by E. F. Griep and C. S. Goddin. Presented at the SAE National Fuels & Lubricants Meeting, Tulsa, Okla., Nov. 7, 1945.

[b] See *Powerplant Engineering*, Vol. 40, March, 1936, pp. 165-166: "Ignition and Combustion of Diesel Fuels," by G. D. Boerlage and J. J. Broeze.

[c] UOP Survey of Foreign Petroleum Literature, by A. D. Petrov and others. Translations Nos. 346, 374, 451, 452, 491, 516-518.

Table A - Effect of Catalytic Cracking on Ignition Properties

Source of Crude	Aniline Point, F		Cetane Number	
	Charging Stock	Recycle Stock	Charging Stock	Recycle Stock
Pennsylvania	158	146	54	52
Texas	141	115	41	34
Midcontinent	159	149	57	54
California	141	120	43	37

from 21 to 24 carbon atoms in the molecule are distinguished by a combination of very high ignition qualities and unusually low pour points. A second side-chain sharply reduces the ignition properties, but such compounds are still of medium quality for paraffins possessing more than 16 carbon atoms. With regard to ignition quality, they have shown exceptions to the above rule for both paraffins and aromatics. Compounds made up of the benzene ring and rather long paraffinic side chains have medium-range ignition qualities and very low pour points. Because of the enormous number of isomers for hydrocarbons in the diesel fuel range (from 12 to 25 carbon atoms) the field for investigation is practically unlimited.

Experimental Fuels – In his selection of 12 fuels for the study of the effects of cetane number and distillation range, Dr. Landen has made a logical choice. He has used only fuels which may be classed as engine fuels. However, having elected to use practical fuels, he is faced with the difficulty which confronts every experimentalist in this field and one which has been pointed out by Blackwood and Cloud,[4] namely, several variables must be changed at one time. In order to emphasize the importance of this difficulty, I have plotted points for these 12 fuels on a skeleton chart (Fig. A) reproduced from the original by Blackwood and Cloud. While changing the boiling range with constant cetane value, or vice versa, it necessarily follows that viscosity, gravity, and heating value are all changed simultaneously. So the question arises, "What has been the effect of these additional changes?"

Heating value of the fuel is most important in the matter of specific fuel consumption because, other engine conditions being constant, most diesel engines show a surprising constancy in their thermal efficiency with change in fuel. This statement has been found to hold, even in cases where a slow-burning fuel was still burning when the exhaust valve opened. A compensating effect for this loss to the exhaust may be that a higher combustion temperature, and hence higher radiation loss, may result from the faster burning fuels.

Cetane Number – Recently, there has been a growing feeling of dissatisfaction with cetane number as a measure of fuel quality. With all the shortcomings of the method for determining cetane number, and in spite of the fact that it tells us only a small part of what we need to know about the ignition and burning properties of diesel fuels, still it is an indication of a vital property of the fuel. It will have to serve until we get something better. However, it does correlate well with the ignition lag in a variety of diesel engines. Dr. Landen has pointed out that several combustion factors are interrelated with cetane number. His data support other research data which shows that both the nature of the combustion and its duration are affected by the ignition delay.

The fact that a long delay period is followed by shock-type combustion and an accompanying high flame luminosity leads to the conclusion that knock in the compression-ignition engine is fundamentally the same as in the spark-ignition engine; even though in one case it initiates the main combustion and in the other it seems to terminate combustion. Dr. Fiock[6] has stated an opinion to which I should like to subscribe, and I quote the paragraph in its entirety:

"It is a personal opinion that the obscure processes leading to preignition or compression-ignition, and to detonation, are in some way produced by homogeneous reactions, or more specifically to unidentified products of such reactions, which occur between fuel and oxygen prior to the initiation or to the arrival of flame. Extending this theory a bit further, the intermediates which may be responsible for preignition are probably different from those which lead to detonation. Both may be so shortlived that they do not exist at all as stable compounds, and hence may be difficult to identify, even by spectroscopic methods."

To this, let me add that I believe we have been led astray in the belief that detonation in a spark-ignition engine is the direct opposite of fuel knock in the compression-ignition engine, because we have failed to keep in mind that the time factor is important and that the direction of its control is the thing that is opposite in the two types of engines. With our present limited knowledge, we must not allow sufficient time for the chain reaction to go on to its ultimate conclusion in the explosive type of combustion if we are to avoid knock in either type of engine. In either engine it is the preflame reactions that we seek to control. In spark-ignition combustion we fight a delaying action to hold the unburned material until it can be reached and burned by the normal progression of the flame to the last bit of combustible material. In compression-ignition combustion, a similar delay is undesirable because, during the period of ignition lag, there is no advancing flame front. On the contrary, the igniting action must be hastened in order to ignite and burn the fuel before it has time for the chain reaction to go on to its conclusion. Under conditions causing extremely long ignition delay, an engine will knock just as hard on high-cetane fuel as it is possible to knock within the limits of engine destruction.

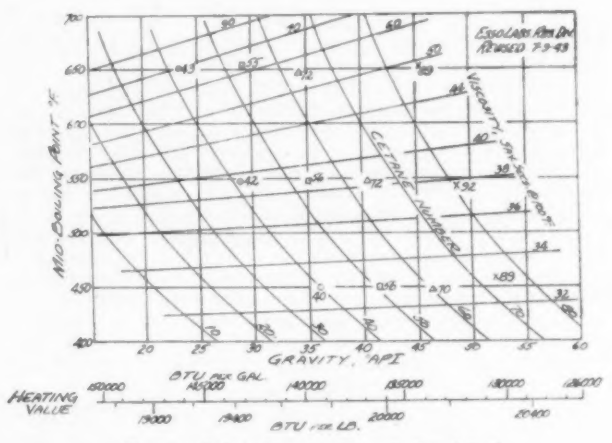
There is much talk of improving engine operation by the use of an extremely high-cetane fuel. The answer will not be forthcoming until someone develops an engine to take full advantage of such fuel[8]. While there are many considerations which would lead us to expect unfavorable results, any such conclusions would be founded upon data taken in engines designed to run on midcetane fuel and involving a very limited range of types of high-cetane fuel.

Paraffinicity Factors – Dr. Landen's Fig. 4 shows a straight-line relationship between cetane number and the per cent paraffin in straight-run fuels. Except for the difficulty of making an accurate determination of the per cent paraffin in a fuel, this would furnish a good indication of its ignition properties. It is estimated that the probable accuracy of the paraffin content of the fuels for which Fig. 4 was plotted is $\pm 3\%$. The UOP characterization factor, which is an indication of the paraffinicity of a petroleum product, is derived from data for the boiling range and the density of the product. It has been correlated with a large number of straight-run distillate gas oils with respect to their breakdown into hydrocarbon types. The breakdown analysis of these gas oils was made by determining the refractive indexes and the densities. Aniline point is also an indication of paraffinicity. Even though such values are only approximations and may be in error so far as paraffin content is concerned, they seem to show surprisingly close correlation with ignition quality.

Flame Initiation and Smoke Studies – There is good agreement between Dr. Landen's results showing the effect of ignition lag on the luminosity of combustion, the formation of carbon during combustion, and the duration of combustion with Mr. Miller's[7] slow-motion study of diesel combustion. Dr. Landen's results, indicating that high-cetane fuels are slow burners, agree with the findings of Robertson et al.[5]

The indication that late injection will cause increased smoke formation is contrary to Mr. Miller's[7] pictures, which show that too early injection will cause an increased amount of smoke. It would seem that there is a best fuel-injection timing for any combination of fuel and engine, and that a shift in either direction from this optimum timing could result in an increase in the amount of smoke. However, it must be kept in mind that exhaust smoke from an engine may be different from that appearing in the combustion chamber of a special test engine during the process of combustion.

It seems that there are many ways in which an engine can produce smoke, and there is more than one type of smoke. To dis-



■ Fig. A – Related properties of diesel fuels

[4] See SAE Journal (Transactions), Vol. 46, February, 1940, pp. 49-53: "Characteristics of Diesel Fuels Influencing Power and Economy," by A. J. Blackwood and G. H. Cloud.

[5] See SAE Journal (Transactions), Vol. 49, August, 1941, pp. 332-338 + (disc.) 338-339: "Present Status of Combustion Research," by E. F. Fiock.

[7] See SAE Journal (Transactions), Vol. 53, December, 1945, pp. 719-733 + (disc.) 733-735: "Slow-Motion Study of Injection and Combustion of Fuel in Diesel Engine," by C. D. Miller.

tinguish the conditions for making any particular type of smoke, visual observation adds much to the information gained from the smokemeter determination. Dr. Landen has made a valuable contribution to the study of the causes of smoke in diesel-engine exhaust.

Dr. Landen's Fig. 17 shows the definite relation between cetane number and ignition delay for an engine operating at 1800 rpm. The fact that the curves for the two compression ratios are parallel confirms the statement that any fuel has to wait during the compression stroke until temperature and pressure conditions have risen enough to start the action. In this case, the additional crank movement for the 12 to 1 ratio over the 15 to 1 ratio is about 3 deg. A detail in the drawing of the curve for 15 to 1 compression ratio indicates an ignition lag of 0.8 millisec. If this figure is intended, it goes lower than anything that has been shown for engines operating at somewhat lower speeds. Minimum ignition lags of slightly under one millisec were found by Wilson and Rose[g] by a preponderance of data from an open-head 15.6 to 1 compression ratio engine operating at 1200 rpm. Schweitzer[h] set up equipment and engine to determine if the statement of a minimum value of ignition lag could hold true under extreme conditions. For pure cetane at a compression ratio of 24 to 1, an air intake temperature of 200 F, 9 in. of Hg supercharge, and 900 rpm, an ignition lag of 0.89 millisec was obtained, which he concluded was as close to the asymptotic limit as the ignition lag can attain under any circumstances.

Dr. Landen has an excellent arrangement for getting two pressure traces on the same film. His arrangement for continuous calibration of the quartz crystal pickup has overcome one of the objections to this type of pickup.

The subject of combustion research as it is related to diesel engines and their fuels furnishes a broad field for investigation, and one that offers almost unlimited possibilities for extending the utility of the diesel engine. The type of work that Dr. Landen is doing should be extended and some effort made to bring about better coordination of effort between the several laboratories working on such projects.

Questions – In closing, the following questions are listed in the hope that Dr. Landen has some further explanation that will be helpful:

1. What assurance is there that the carbon which appears during combustion is the result of cracking rather than a case of hydrogen burning faster than the carbon?

2. Can any distinction be made between the types of smoke for different operating conditions, for example:

(a) Too early versus too late timing of fuel injection?

(b) Long ignition lag versus short ignition lag?

(c) High-cetane fuel versus low-cetane fuel?

3. What are some of the extreme rates of pressure rise (Table 3) in terms of pounds per square inch per degree of crank angle and what is the correlation between these rates of pressure rise and the audible knock?

Acknowledgment – The author gratefully acknowledges the helpful assistance of J. S. Bogen, R. M. Nichols, and Dorothy E. Fisher in the preparation of this discussion.

Another Explanation Offered For Diesel-Engine Exhaust Smoke

—P. H. SCHWEITZER

Pennsylvania State College

THE author, with other observers, has noticed that under certain circumstances, the density of the exhaust smoke increases as the cetane number of the fuel increases, and puts forward the theory that paraffinic high-cetane fuels produce more carbon in early stages of combustion than the low-cetane fuels and, hence, take longer for the carbon to be consumed in the cylinder. This, he claims, may explain the longer flame for the higher cetane fuels and causes more black smoke. The greater amount of carbon produced in burning the thermally least stable paraffins is caused, according to the author, by the pyrolytic action in the combustion chamber. This cracking process precipitates the carbon.

The writer disagrees with this explanation of the smoke frequently observed in burning high-cetane fuels. High-cetane fuels that have very short ignition lag are smoky irrespective of their chemical composition. For instance, doped aromatics, whose cetane has been brought up by additions to, say, 70 cetane, will have smoky exhaust unless it is burnt at a very low compression ratio. The author himself remarks, referring to his Fig. 15, that the 70-cetane blended fuel (with cetane additions) gave smoke readings similar to the 70-cetane straight-run fuel. The only time when the cetane improver reduced the smoke was when, at the same time, it made the fuel more volatile, when, for instance, to a 600-700 F mid-boiling point fuel a 380 F mid-boiling point cetane improver was added. The improvement in volatility was then so great that the upping of the cetane could not quite spoil it as far as exhaust smoke is concerned.

It is the opinion of the writer that the effect of ignition quality on exhaust smoke can be explained without pyrolytic action. If the ignition lag is either too short or too long, we have smoke in the exhaust. Too short an ignition lag either prevents the fuel drops from penetrating far enough to meet oxygen-rich air before it ignites, or fails, upon ignition, to produce a sufficiently intense turbulence to ensure intimate mixing. The important thing is that too short an ignition lag will be accompanied by exhaust smoke irrespective of whether it was produced by high-cetane paraffinic fuel, high-cetane nonparaffinic (doped) fuel, extra high compression ratio, extra high inlet air temperature, or a combination of these factors. The physical factor of ignition lag will control the exhaust smoke, rather than the chemical composition of the fuel.

We also get increased exhaust smoke if the ignition lag is too long, because combustion cannot be completed when the exhaust opens. By making the ignition lag longer and longer, a point is reached where the engine misfires. In the border region of misfiring, the engine always smokes. Everything that will make the engine misfire, like too low compression or too low cetane, will, in a smaller dose, make the engine smoke. Incidentally, this explains the smoke at very light load.

So, we encounter smoke at both ends of the cetane scale, but it is not possible to state where those ends are. Ignition quality is only one factor in the smoke and not the most important factor. Ignition lag refers to the fuel particle which ignites first, exhaust smoke is caused by the fuel particles that ignite last or next to last. In other words, it is the duration of the combustion that counts.

The duration of the combustion is controlled not so much by the ignition lag (although short ignition lag tends to make it longer) as by two other factors: by the length of injection and by the air-fuel ratio of the spots where the fuel particles burn. Too long injection and too rich mixture cause smoke. When the load is increased we encounter both, the injection period is lengthened and the air-fuel ratio is decreased. Good turbulence improves the local air-fuel ratio of the spots where without it the mixture would be too rich. Finally, high fuel volatility promotes the mixing of fuel with air and, therefore, reduces smoke.

The author mentioned that the high compression ratio engine had a slightly higher rate of pressure rise. This seems to be out of line with the notion that high compression ratio makes for short ignition lag and smoother burning. But what the author has measured is really the sum total of the compression pressure rise and combustion pressure rise. If the ignition is early, the compression pressure rise is appreciable and it is higher with high compression ratio. Therefore, the increase in the rate of compression pressure rise may exceed the decrease in the rate of combustion pressure rise. With ignition at or close to top dead center, that would not be true.

The use of a strobotron for marking the degree scale of a cathode-ray indicator is ingenious. The very short ignition lags shown in Fig. 17 are doubtful.

Discusses Part II Of Author's Paper

—PHIL MYERS and OTTO UYEHARA

University of Wisconsin

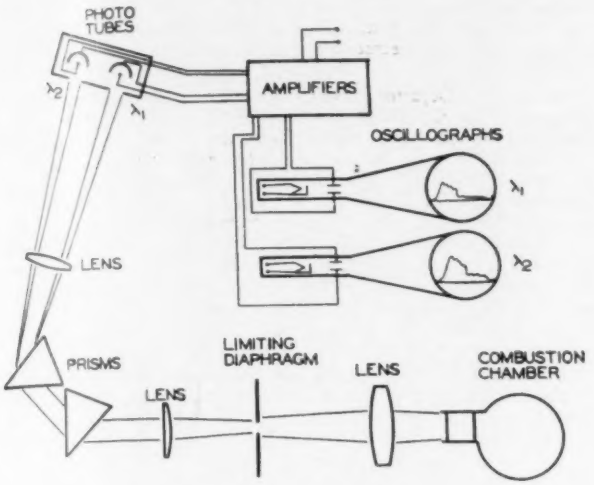
IN studies made at the University of Wisconsin the writers have observed that the flame in a spark-ignition engine during detonation bears a marked resemblance to the luminous flame of the diesel. If this similarity has any significance, it could well be that the mechanism of detonation in the spark-ignition engine is similar to the mechanism of combustion in the diesel.

The author, in his discussion of the order of events in diesel combustion, mentions temperatures ranging from 1000 to 2000 F higher than the temperature of the compressed gases. A word of explanation as to how these temperatures were determined would be appreciated. It should also be emphasized that the reaction rates quoted are cracking rates, not oxidation rates, and should not be confused with the minimum ignition lag of 0.001 sec as reported by Wilson and Rose.[g]

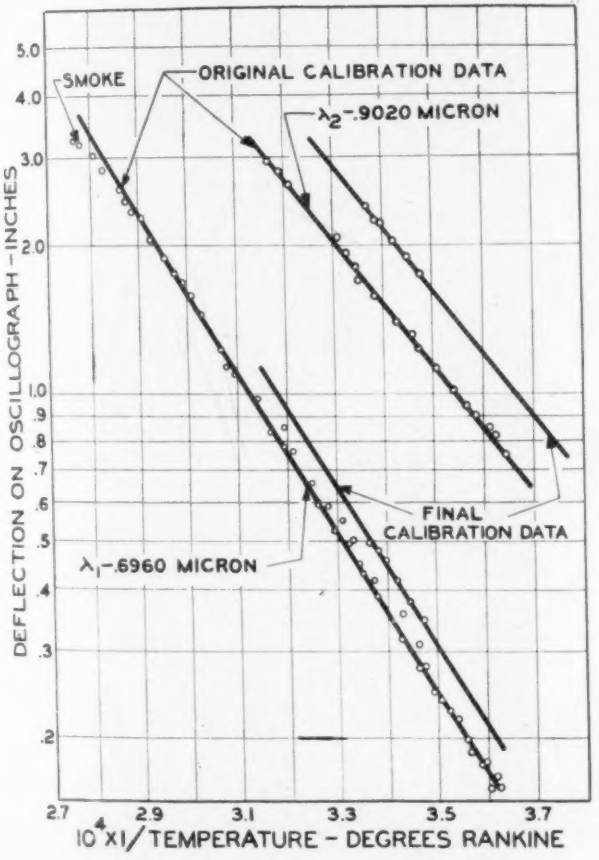
The writers are keenly interested in the photoelectric pyrometer described by the author because they have been using a similar in-

[g] See SAE Journal (Transactions), Vol. 41, August, 1937, pp. 343-348: "Behavior of High- and Low-Cetane Diesel Fuels," by G. C. Wilson and R. A. Rose.

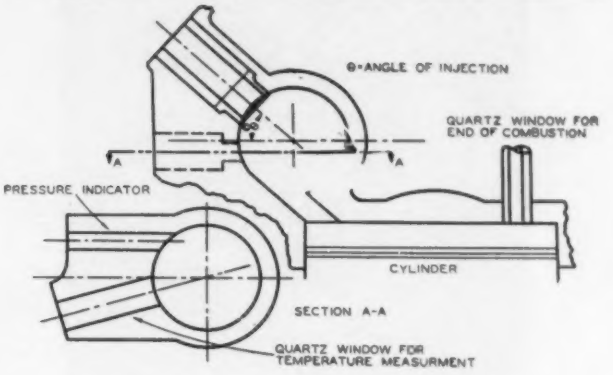
[h] See *Automotive Industries*, Vol. 78, June 15, 1938, pp. 848, 854: "Injection of Diesel Fuel into Flame Cuts Ignition Lag Only Moderately," by P. H. Schweitzer.



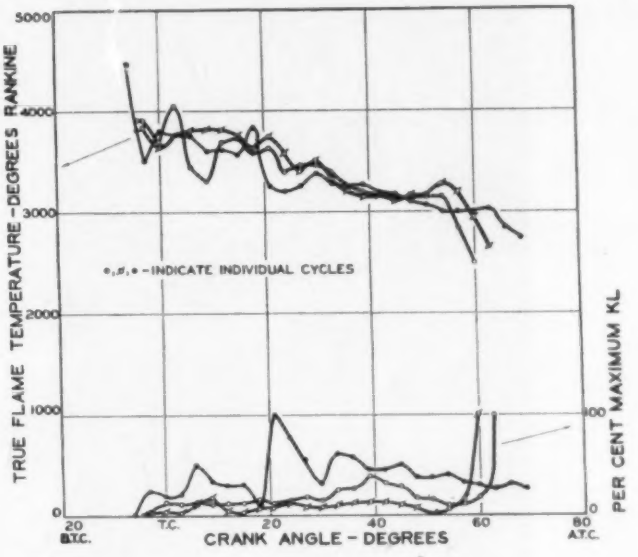
■ Fig. A. – Schematic diagram of electro-optical pyromter



■ Fig. B – Black-body calibration data



■ Fig. C – Schematic drawing of engine head



■ Fig. D – Temperature and KL fluctuations under constant operating conditions

Table A – Determination of True Flame Temperatures

I. First Comparison	
Standard Optical Pyrometer (thin axis of flame):	
Temperature of Lamp through Flame (average of two readings), R	4026
Temperature of Lamp (average of two readings), R	4025
Electro-Optical Pyrometer Readings (thin axis of flame):	
Deflection at $_1$, in.	1.84
Deflection at $_2$, in.	2.84
True Temperature, R	4020
II. Second Comparison	
Standard Optical Pyrometer (thin axis of flame):	
First Trial:	
Temperature of Lamp through Flame (average of three readings), R	4076
Temperature of Lamp, R	4145
Second Trial (lower lamp current):	
Temperature of Lamp through Flame (average of three readings), R	4025
Temperature of lamp, R	4005
Using a graphical interpolation, the temperature of the lamp when sighted through the flame equals the temperature of the lamp at 4035R.	
Electro-Optical Pyrometer Readings (thin axis of flame):	
Deflection at $_1$, in.	1.86
Deflection at $_2$, in.	2.85
True Temperature, R	4020

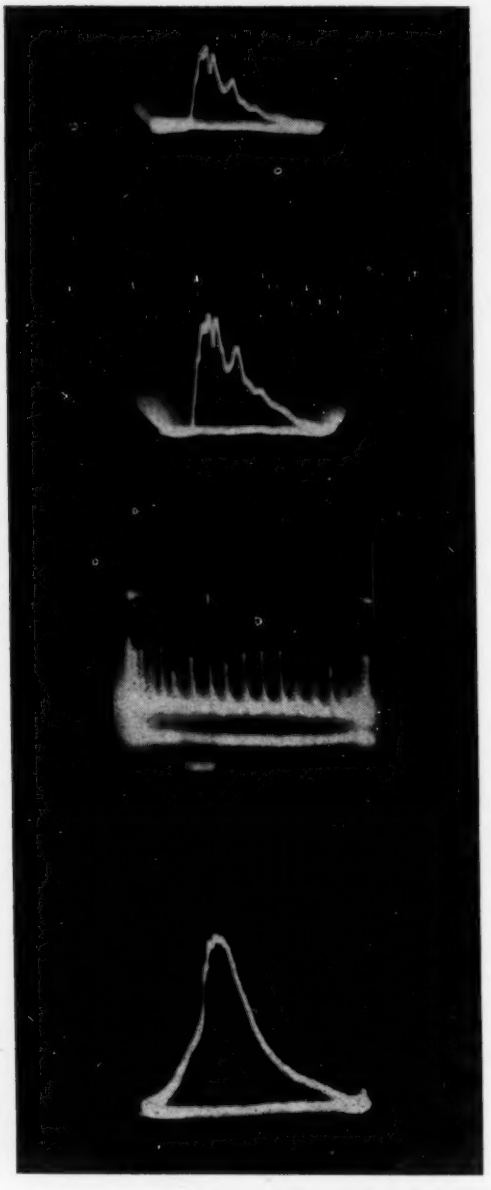
Table B – Electro-Optical Pyrometer Readings

(Thick axis of flame)

Deflection at $_1$, in.	2.3
Deflection at $_2$, in.	3.6
True Temperature, R	402

strument in diesel combustion research at the University of Wisconsin for some time. If the author is free to reveal such information, it would be of interest to know the effective wavelength values for the red and green radiations indicated in Fig. 20, the range of wavelengths covered, and the reason for choosing these particular wavelengths.

The continuous emission spectra of the diesel flame is to be expected because of its luminosity. The statement that the primary source of radiation is the incandescent carbon particles is undoubt-



■ Fig. E – Oscillogram of entire combustion process with injection at 24 deg BTC

edly correct. This continuous radiation is the basis for the design of the electro-optical pyrometer[17] developed at the University of Wisconsin.

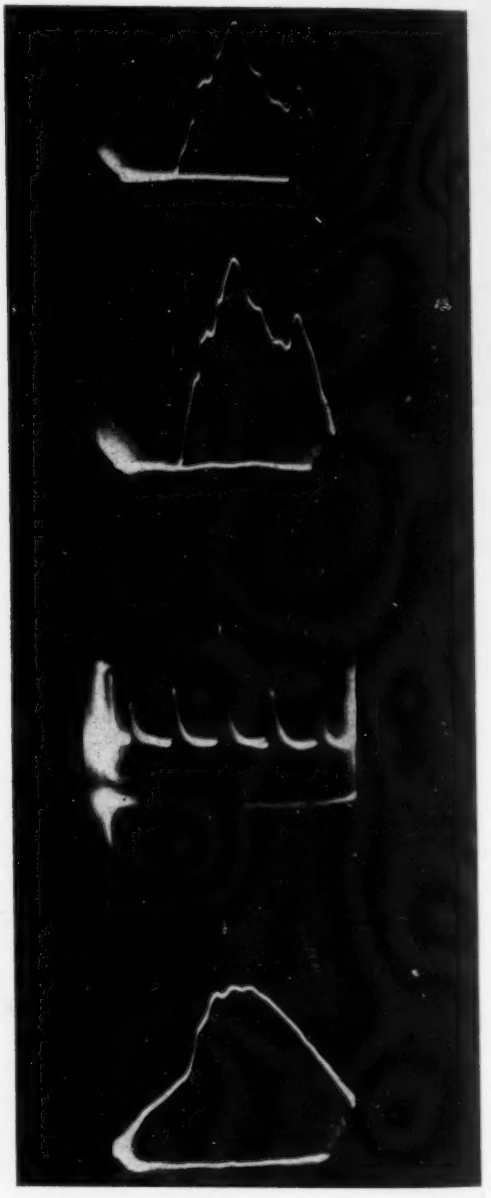
The reason for the change in the prominence of the OH band with the change in combustion-chamber design is not clear. However, experiments at the University of Wisconsin have shown a considerable change in flame luminosity with a change in operating conditions. The apparent increase in OH intensity might actually result from a decrease in the intensity of the continuous background radiation.

The author's conclusion that the continuous absorption at 2 deg btc is caused by carbon precipitate is confirmed by the radiation curves shown in Fig. 20. It is noticeable there that the beginning of combustion is characterized by a rapid increase in radiation intensity which would correspond to the carbon deposition. Spectrograms taken in the visible and infrared portion of the spectrum should help to establish the character of this absorption.

The above data, and some of the seeming inconsistencies brought out in Part I of the paper and in Mr. Wilson's discussion, point to the need for extensive fundamental research on the diesel combustion process. This need has been emphasized recently by Griep and Goddin[8] who have pointed out that basic combustion studies of hydrocarbons in the diesel engine are at about the same level now as were combustion studies on the spark-ignition engine in the 1920's. It is further pointed out that fundamental data are needed not only on the ignition delay of fuels but also on their rate of burning, since this characteristic has a marked influence on efficiency and engine wear. Data of this nature should also be of interest and value in connection with gas turbine development and operation. The author's instrumentation appears to be directly applicable to this problem.

In 1942 the Departments of Chemical Engineering and Mechanical Engineering at the University of Wisconsin jointly initiated a diesel combustion research program, the objective of which was to obtain complete combustion data on a diesel engine, that is, to determine the influence of different factors by systematically varying the engine operating conditions, the fuel composition, and the combustion-chamber design. A report of the instrumentation developed in connection with this program has recently been published.[17] The writers have employed an electro-optical pyrometer in their research studies for approximately a year and a half, and possibly a brief résumé of some of their experiences in the development and use of this instrument may not be out of place in this discussion of the same type of instrument described by the author.

A schematic diagram of the electro-optical pyrometer developed at the University of Wisconsin is shown in Fig. A. The radiation from the combustion chamber is focused by a lens system on a limiting diaphragm which has an opening somewhat smaller than



■ Fig. F – Oscillogram with injection at 24 deg BTC and with horizontal scale expanded

■ Fig. G – Oscillogram with injection at 24 deg BTC and with horizontal scale expanded

the image of the source of radiation. Thus, vibration and other small movements of the source of radiation do not cause a variation in the light permitted to pass the limiting diaphragm. After the radiation passes through the diaphragm it is paralleled by a lens, dispersed by two prisms, and again focused by another lens. A barrier containing two slits is situated at the focal point of this lens. A phototube is located behind each slit in such a position that it is actuated by the radiation passing through the slit. The outputs from the phototubes are amplified and impressed on the vertical plates of the single-beam cathode-ray tubes. In this particular application the cathode rays are displaced horizontally in proportion to the engine crank position. In this manner, a graph of monochromatic radiation intensity or apparent temperature as ordinate and crank angle as abscissa is repeatedly traced on each cathode-ray screen.

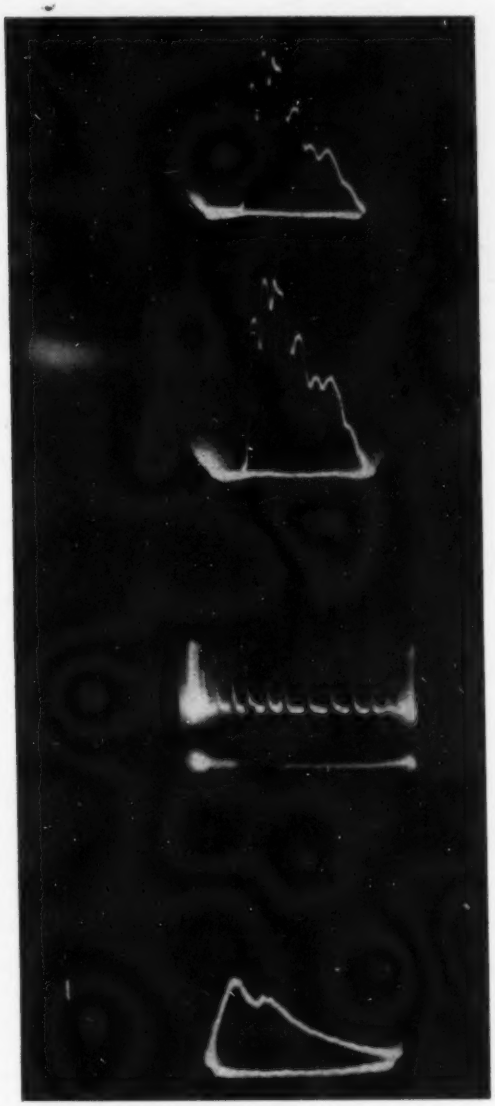
A black body was used in calibrating this electro-optical pyrometer. Black-body conditions were established by drilling a hole with a conical bottom in a graphite block. The graphite block was heated in the crucible of an induction furnace and an optical pyrometer with a lamp calibrated by the Bureau of Standards was used to determine the temperature of the black body. The graphite block was blanketed with nitrogen to prevent smoke formation. The calibration data are shown in Fig. B. Immediately after completing this calibration, a broad-ribbon tungsten lamp was calibrated against the electro-optical pyrometer for use as a secondary standard.

The fundamental accuracy of the method and calibration was verified by setting up a steady, high-temperature flame so that its temperature could be measured by the electro-optical pyrometer and by a standard method. The experimental flame, using acetylene, oxygen, and air, had a cross-section approximately ¼ in. by 1 in. and was set up inside a water-cooled jacket to simulate engine conditions. Two peep holes were provided through which observations would be made. The results obtained with this apparatus are shown in Table A. The effect of flame thickness (with the burner rotated 90 deg.) is shown in Table B.

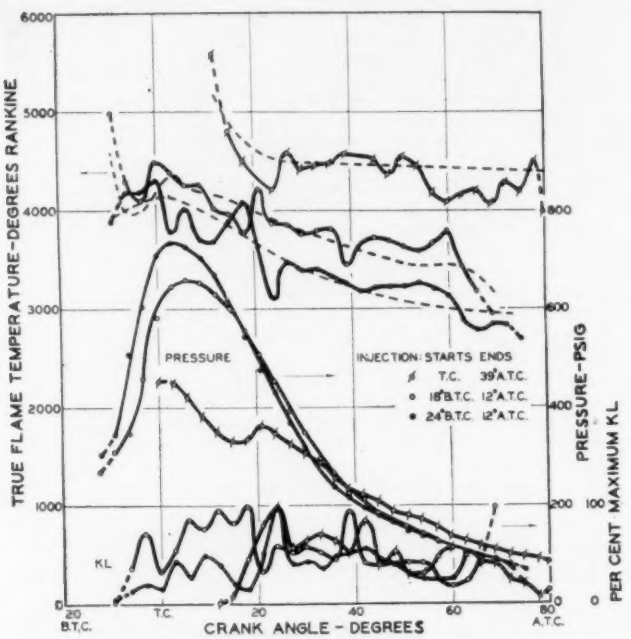
The engine with which the electro-optical pyrometer is being used is of the precombustion-chamber type with openings into the head as shown in Fig. C. In order to determine which opening would be best suited for temperature measurements, quartz windows were installed and observations were made by alternately sighting the electro-optical pyrometer through the two openings in the precombustion chamber with the engine running under constant operating conditions.

The temperatures determined through the tangential window were found to be slightly lower than those determined through the radial window, particularly during the first part of combustion. This was undoubtedly a result of the cooling effect of the combustion-chamber walls. During the last half of combustion there was very little difference in the two temperatures. No appreciable difference in ignition lag measurements through the two openings could be detected. The radial opening was finally selected for subsequent temperature measurements.

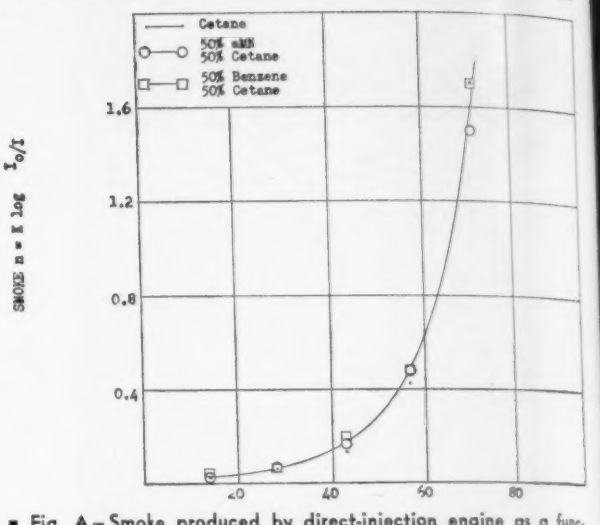
The results of an investigation of engine cyclic reproducibility



■ Fig. H – Oscillogram of whole combustion process with injection at top center



■ Fig. I – True temperature, pressure, and KL fluctuations when point of injection is varied



■ Fig. A – Smoke produced by direct-injection engine as a function of bmep – fuels used were cetane and blends of cetane with alpha methyl naphthalene and benzene

under constant operating conditions are shown in Fig. D. These data show that the average trend is reproduced, but the temperature at any particular point is not necessarily reproduced from cycle to cycle.

Oscillograms of the combustion process are shown in Figs. E, F, G, and H. Figures E and H illustrate the effect of changing the point of injection. The two top traces are the apparent temperatures, the next trace shows the timing marks, the next the injection, and the bottom trace the pressure. The horizontal scale in all cases is engine crank angle. The timing marks are 12 deg apart, except in Fig. G where they are 4½ deg apart. The high timing mark indicates top center in all cases.

The true temperature, the pressure, and the *KL* factor fluctuations, as determined from the oscillograms in Figs. E and H, are shown by the solid lines in Fig. I, together with the data for a single cycle with injection starting at 18 deg before top center. Each of the dotted lines in Fig. I was obtained by analyzing 10 cycles and drawing average curves for these data. The trends shown have been confirmed by check runs.

The writers wish to emphasize again the need for fundamental combustion data such as may be obtained with the instrumentation described by the author of the paper under discussion. It is hoped that results to be obtained with this apparatus will be published in the near future.

Author's Closure To Discussions

IF Mr. Wilson means that the hydrogen is burning from the hydrocarbon molecule leaving a carbon residue, one would expect this to be gradual. However, the absorption spectra in Fig. 27 show a very heavy and rapid deposit. Simple hydrocarbons react with oxygen to form aldehydes, and in fact aldehydes have been shown to be present in engines. Thus, the molecule must undergo intermediate reactions and the oxygen does not attack the hydrogen of the molecule as such. Bond energy of C-H is given a higher value in the literature than the bond energy of C-C, and from this point of view the molecule might break more easily than the hydrogen stripping off directly. When the oxygen is not available it is known that the hydrocarbon reacts to form cracked products as well as polymerized products. Any hydrogen liberated during cracking could cause a vigorous burning upon coming in contact with the oxygen in the cylinder.

Distinction of smoke types is very difficult and visual observations can often be misleading. To answer, specific data must be given and these data would have to include spectral distribution of the smoke. The high-boiling, low-cetane fuel gave a blue smoke at low bmep. As the cetane increased, the smoke approached a black variety to a degree. All the lower boiling fuels gave a black variety of smoke.

Audible knock seemed to follow the rates of pressure rise very closely.

Rates of pressure rise in psi per deg crank angle are as follows:

Compression Ratio	Injection Start, deg BTC	Cetane 40	55	70	90
12 to 1	19	108	73	60	43
12 to 1	29	185	132	99	71
15 to 1	19	109	75	63	52
15 to 1	29	207	142	109	91

In reply to the queries by Messrs. Myers and Uyehara about the temperatures 1000 F to 2000 F being higher than the surrounding gases, I would say that the photograph of fuel droplets reveals incandescent regions. The incandescent material would have to be of much higher temperature than the surrounding gases, and hence this statement. Recent, slow-motion studies of injection and combustion of fuel in a diesel engine made by C. D. Miller, NACA, show localized regions of incandescence before the charge is totally inflamed.[f]

Our photoelectric pyrometer at present has the wavelength settings of 8000 and 5020 A. These values were chosen because they are well out of range of emission due to water and a copper arc which served as a convenient calibrating source for these wavelengths. The wavelength band selected by the monochrometer has not yet been determined.

It would be illuminating if someone would take some high-speed photographs by an absorption method. By placing a suitable light source of sufficient intensity on one side of his combustion chamber and a neutral absorbing filter between the chamber and camera, such a picture could be obtained. Light from the combustion itself would thereby be largely absorbed, permitting the light from the source to indicate absorbing material within the chamber. By this method the smoke formation and consumption within the combustion chamber could be studied during the combustion cycle.

Mr. Schweitzer, in his discussion, gives a number of statements concerning smoke and the physical factors which lead to smoke, but does not attempt an explanation of smoke formation. When a liquid hydrocarbon is injected into the combustion chamber of an engine and carbon comes out in the form of smoke, one must concede that some reactions have occurred which precipitate soot. The author has presented an explanation for this smoke production. While smoke is influenced by the physical factors of fuel distribution and mixing it must also be remembered that chemical reactions are involved in converting liquid fuel to the end products. The entire smoke picture has not been worked out very satisfactorily because of lack of agreement between measuring techniques in different laboratories. The usual practice of sampling exhaust gases and measuring the smoke is not reliable when comparing smoke from different engines, or changing operating conditions. Volume drawn through the measuring tube, temperature of the gas as it passes through the tube, engine speed, displacement, and resonance effects in the measuring

concluded on page 296

Gasoline Gum Tolerance of Ordnance Equipment

(Compiled from Reports of Coordinating Research Council, Inc.)

by M. L. ALSPAUGH
Ethyl Corp.

COMMERCIAL motor gasolines, when subjected to long periods of storage, particularly at high ambient temperatures, may undergo deterioration as measured by an increase in gum content. Although gasolines meeting U. S. Army specifications 2-103B have an unusually high storage stability, they are distributed throughout many theaters of operation where time of storage and the exposure to high atmospheric temperatures may represent extremes not encountered in ordinary commercial operations. The deterioration of these stocks is of particular concern in military operations. Thus, it is possible to foresee situations in which it may be necessary to operate military vehicles on gasolines having higher gum contents than permitted by the specification.

In view of this possibility, the Coordinating Research Council was requested by the Ordnance Department to recommend the maximum permissible limits of ASTM gum in aged fuels that could be used safely in Ordnance equipment during emergencies. Little information was readily available on which to base a recommendation. However, certain limits, based on meager commercial experience, were set arbitrarily for the guidance of field officers in using these fuels until field data could be obtained.

The Gasoline Additives Group of the CFR Motor Fuels Division undertook to obtain the necessary field data in cooperation with the Ordnance Department. They set up and supervised three field tests on Ordnance equipment. Two tests were conducted on wheeled and track-laying vehicles at Camp Bullis, San Antonio, Tex., during the periods of Oct. 15, 1944, to Jan. 15, 1945, and April 16, 1945, to June 1, 1945, respectively. The third test was conducted on Ordnance stationary engines at San Bernardino, Calif., from Aug. 9, 1945, to Sept. 8, 1945.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 11, 1946.]

WHEN subjected to long storage periods, particularly at high temperatures, gasoline may undergo an increase in gum content.

For this reason the Ordnance Department requested CRC to recommend the maximum permissible limits of ASTM gum in aged fuels that could safely be used during emergencies.

The tests reported in this paper were undertaken by the Gasoline Additives Group of the CFR Motor Fuels Division.

THE AUTHOR: M. L. ALSPAUGH has been doing experimental work on fuels and engines at Ethyl Corp. for the past four years. He previously had been an assistant at Ethyl's testing laboratory in Tulsa, and from the period from 1929-1936, he was with Mid-Continent Petroleum Corp. Mr. Alspaugh is a graduate of Oklahoma A & M.

It should be pointed out that the objective of these tests was to obtain data on the effect of fuels containing preformed gum on the deposits found in the induction and fuel systems of the various types of vehicles in order to determine the maximum limit of ASTM gum in aged fuels which could be used safely in cases of emergencies. It is not the intention of this paper to criticize in any way the performance of any type of engine regardless of its operation on fuels of high gum content.

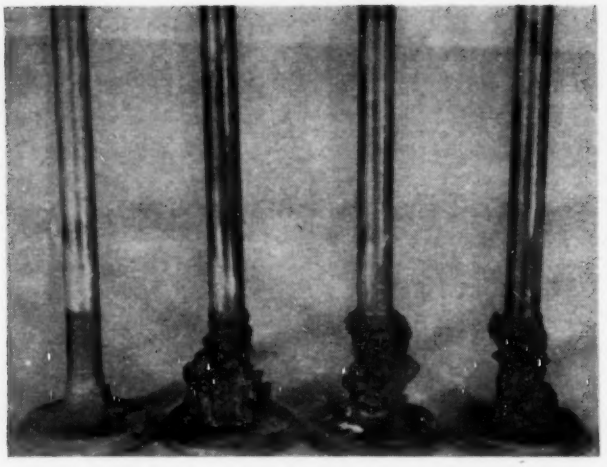
For the purposes of this paper, the term "gum" is defined as the "ASTM gum per 100 ml of gasoline."

■ Wheeled and Track-Laying Equipment Tests

Fuels – The fuels used in the vehicle tests Nos. 1 and 2 are shown in Table 1. The high-gum fuels in the first test were made by blending 2-103B specification gasoline with a fuel containing approximately 900 mg of gum. This high-gum fuel was an uninhibited, naturally aged, cracked stock. The high-gum fuels for the second test were made by blending 2-103B specification gasoline with a naturally aged, cracked stock containing 30 mg of gum. To make the 50-mg gum fuel, it was necessary to add approximately 5% of the 900-mg gum fuel from the first test.

It should be pointed out that none of these test fuels was naturally aged 2-103B specification fuels, but the fuels used were the best compromises that could be made under the existing conditions. The validity of the results of the first test was questioned, owing to the way in which the high-gum fuels were made. It was thought that inasmuch as the 900-mg gum fuel was a special fuel, the blended high-gum fuels probably were not comparable to the fuels encountered in the field. For this reason it was decided to use the 30-mg gum stock to make as many high-gum fuels as possible for the second test. Some complications resulted from this decision because the octane number of the 30-mg gum fuel was below the specification for 2-103B gasoline. It was necessary, therefore, to blend high-octane stock with this fuel to raise the octane number to the required level. It also seemed advisable to check the validity of the first test results by including in the second test a 25-mg ASTM gum fuel made with the 900-mg gum stock.

Fuels for both tests contained 3.0 ml of tetraethyl lead per gal. The fuels used in the first test contained 35 lb of antioxidant per 1000 bbl, and the fuels in the second test contained 10 lb of antioxidant per 1000 bbl. Lubricating oil meeting 2-104B was used throughout the tests.



■ Fig. 1 – Intake valves for vehicle *A* showing relative amount of deposits – mg of gum in fuel (reading from left to right): 1.5, 10, 25, 50

Vehicles – The vehicles used in the tests were those recommended by the Ordnance Department as being representative of the types found in the field:

Wheeled Vehicles	Combat Vehicles
Truck, ¼-ton 4x4	M-24 Light Tank
Truck, ¾-ton 4x4	M4A3E8 Medium Tank
Truck, 2½-ton 6x6	M-18 Gun Motor Carriage
Truck, 4-ton 6x6	

The number and types of vehicles operated on each of the test fuels are shown in Table 2.

In addition, one vehicle during the first test was placed on a simulated dead storage test using the 25-mg gum fuel. In the second test, three vehicles were placed on a simulated dead storage test, one on 25-mg gum fuel (second test), one on 25-mg gum fuel (first test), and one on 50-mg gum fuel. These simulated dead storage tests were for the duration of the road tests, approximately 45 days in the first test and 30 days in the second test.

All vehicles for both tests were equipped with new engines and fuel systems. After a break-in run of 500 miles for the wheeled vehicles and 50 miles for the track-laying vehicles, a complete inspection of the fuel and induction system of each vehicle was made to ensure that all parts were clean and free of deposits. Also, at this time the engine of each vehicle was adjusted in accordance with the recommended Ordnance Department procedure (ignition timing, valve clearance, and so on). The wheeled vehicles carried a 100% overload and the track-laying vehicles a normal combat load.

Test Course and Schedule – The test course was the same for both tests. For the wheeled vehicles it included both highway and cross-country operation. The highway operation was over a paved course 21 miles long and was such as would be encountered in normal driving. The cross-country course was over a hill route approximately 2 miles long. The grade on the up side of the hill was approximately 20% and on the descending side was approximately 26%. The track-laying vehicles operated over a course approximately 20 miles long, 13 miles of which was over packed gravel roads and 7 miles over a cross-country route. All vehicles operated as much of the time as possible at their recommended maximum engine speed and were driven in such a gear as to hold their road speed slightly below the recommended maximum.

The test fleet was operated in three groups, the wheeled vehicles in two groups and the combat vehicles as one group. A definite test schedule was laid out for both the wheeled and track-laying vehicles. It involved periods of part- and full-throttle operation, high- and low-speed driving, idling and cooling periods of 15 min to 24 hr. The vehicles on the simulated dead storage test were run on the highway for approximately 2 miles each week and then returned to storage.

Daily records were kept on the weather conditions and on the starting and warmup characteristics, running performance, mileage, gasoline and oil consumption of each of the vehicles. Inspections of fuel and induction systems were made and recorded each week to determine if any gum deposits were being formed. Standard Army preventive maintenance was practiced, and records were kept of all maintenance work required for each vehicle.

The duration of the first test was 5000 miles for the wheeled vehicles and 100 road hr for the track-laying vehicles. The duration of the second test was 10,000 miles for wheeled vehicles and 150 road hr for track-laying vehicles. At the completion of each of the tests the engines and fuel systems were disassembled completely for inspection and significant parts were photographed.

Summary of Results – First test: During the first test, the atmospheric temperatures averaged about 50 F and never exceeded 70 F. Examination of the engine parts indicated that for all engines tested only slight differences were obtained in the amount of deposits as the ASTM gum content of the fuel increased. The vehicles operating on the 1.5-mg gum fuel were exceptionally clean, but even

Table 1 – Inspection Data

	First Test			Second Test				
ASTM Gum Content, mg per 100 ml	1.5	10	25	1.5	10	25	50	25[b]
Gravity, deg API	57.5	57.2	57.0	61.7	60.5	60.2	59.3	60.7
Distillation, ASTM[a]								
IBP, F	109	110	106	110	106	107	104	115
10% Point	145	146	147	147	148	148	149	155
50% Point	227	229	230	213	228	243	214	221
90% Point	335	340	342	299	338	350	355	309
EP	379	386	391	357	388	395	402	383
Reid Vapor Pressure, psi	6.4	6.4	6.4	6.6	6.7	6.8	6.7	6.1
Induction Period, min	625	398	324	800	130	70	65	300
Copper Dish Gum, mg per 100 ml	7.5	76.4	110.7	5.5	121	243	323	71
5-Hr Army Accelerated Gum, mg per 100 ml	34.9	251.8	394.1	14	900	1250	1000	500
Sulfur, %	0.052	0.052	0.050	0.03	0.10	0.14	0.17	0.05
Octane Number								
ASTM Motor Method	81.0	81.0	80.5	80.0	78.5	78.0	77.5	79.0
CFR Research Method	91.0	91.0	90.5	84.5	85.5	86.0	85.5	84.0

[a] Distillation data were based on per cent recovered for first test and per cent evaporated for second test.
[b] Blended as in first test.

the vehicles operating on the 10- and 25-mg gum fuels contained deposit accumulations that would have been ignored or considered normal if inspected alone. There were no significant differences in engine deposits or engine cleanliness between the vehicles operating on 10-mg and 25-mg gum fuels.

Deposition was greatest in the intake ports and on the intake valve tulips; however, none of the vehicles showed enough deposits to indicate that extensive repairs would be necessary, provided the length of operation did not exceed that used in this test (5000 miles for wheeled vehicles and 100 hr for combat vehicles).

The fuel systems of all the vehicles were clean and free of deposits, with the exception that the choke valves of vehicle *C* were stuck in the open position on the engines operating on 25-mg gum fuel and very sluggish on the engines operating on 10-mg gum fuel. However, there were no difficulties in the starting or operation of these vehicles which could be attributed to the choke valves being stuck. This condition existed only during the last 20 hr of the test.

The vehicle on simulated dead storage showed no signs of gum formation in the fuel or induction system.

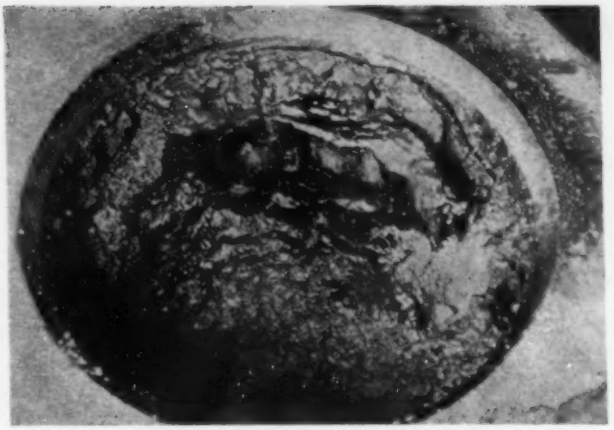
In view of the above results, it was recommended that further tests be made at atmospheric temperatures above 70 F with fuels containing up to 50-mg ASTM gum using a greater variety of vehicles and extended mileages.

Second test: The results of the second test run at atmospheric temperatures above 70 F with additional types of vehicles and at extended mileages differed markedly from those obtained in the first test. In general, they indicated that the use of fuels containing 10-mg ASTM gum or more results in excessive engine deposits that may shorten the duration of satisfactory operation.

Twenty-eight of the 30 vehicles in operation on high-gum fuels finished the test without serious difficulty. Two wheeled vehicles failed before the completion of 10,000 miles, due to intake valve trouble resulting from the accumulation of deposits in the induction system. One of these failures, after approximately 5000 miles in a vehicle operating on 25-mg gum fuel as blended in the first test, was definitely due to gasoline gum. Inspection of this vehicle showed the intake valves had a black sticky deposit on the stems sufficiently heavy to cause the valves to stick. There was some doubt as to whether failure in the other vehicle was actually due to gasoline gum. This vehicle was operating on 50-mg gum fuel and had run for approximately 6500 miles. Inspection of this engine revealed that two of the intake valves were burned badly, and it was assumed that this was the result of deposits on the intake valve tulips and in the intake ports. There was no deposit on the stems of these valves that would cause them to stick. Later comparisons of these intake valves and ports with those of other vehicles of the same type did not show the deposits to be excessive.

Two other vehicles, although they finished the test, experienced difficulty due to gummy deposits with sticking of choke valves and erratic operation of the piston in the vacuum-operated downshift mechanism of the transmission. The two vehicles experiencing this trouble were operated on 25- and 50-mg gum fuels.

Owing to physical limitations, it was not possible to run more than one vehicle of each type on any one fuel. The results, therefore, may not always be specifically applicable to any one type of vehicle and fuel, but they probably are applicable in general. In some cases the wheeled vehicles had more deposits in the intake system when operated on a fuel containing 10 mg of gum than they had with any of the other high-gum fuels. However, a consideration of overall engine cleanliness for the same vehicles indicated that slightly dirtier engines were obtained as the gum content of the fuels increased. The greatest comparative increase in both engine deposits and piston-skirt lacquering was evidenced between fuels with 1.5 and 10 mg gum.



■ Fig. 2 – Intake port of vehicle *A* – 10-mg gum fuel

As a general rule, fuels containing between 10- and 50-mg gum showed very little difference in the amount of induction system deposits. Examination of the various engines operating on the different high-gum fuels showed that the deposits formed had a tendency to become hard and brittle as the gum content of the fuel increased. There was some evidence that deposits resulting from the use of 50-mg fuel were more inclined to flake and break away. This may have had some bearing on the fact that the difference between the amount of deposits from 10- and 50-mg fuel was so small. It was noticed that the point of maximum deposition in the induction systems of the engines varied according to vehicle type. Apparently those vehicles that operated most of the time at nearly full throttle had maximum deposition in the heat riser section and intake manifold branches with lesser amounts on the intake valve tulips and in the intake ports. On the other hand, those vehicles operating a relatively small part of the time at nearly full throttle had maximum deposition in the intake ports and on the intake valve tulips.

The results of this test indicated that wheeled vehicles may be operated on fuels containing 10-mg gum or more for periods up to 5000 miles. After 5000 miles, engine

Table 2 – Number of Vehicles Used with Each Fuel

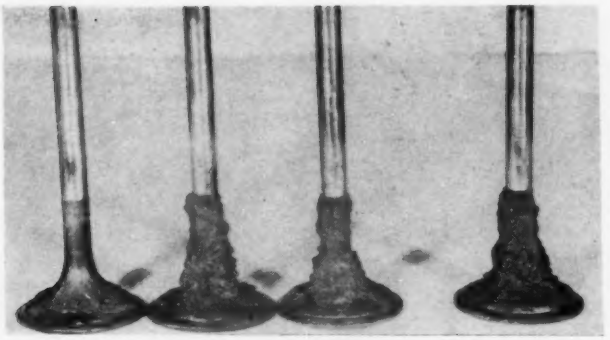
First Test

Fuel, mg	Vehicle *A*	*B*	*C*
1.5	1	1	1
10	2	2	2
25	1	1	1

Second Test

Fuel, mg	Vehicle *A*	*B*	*C*	*D*	*E*	*F*	*G*
1.5	1	1	1	1	1	1	1
10	1	1	1		1	1	1
25	1	1	1	1	1	1	1
50	1	1	1	1	1	1	1
25[a]	1	1	1				

[a] Blended as in first test.



■ Fig. 3 – Intake valves for vehicle *B* showing relative amount of deposits – mg of gum in fuel (reading from left to right): 1.5, 10, 25, 25

failure may be expected in any vehicle using high-gum fuels regardless of whether the gum content is 10 or 50 mg. This condition is probably due to the fact that all fuels containing 10-mg gum or more produce a degree of deposition sufficient to place at the point of incipient failure all vehicles operated on them for at least 5000 miles.

Owing to the fact that the operation of vehicles on fuels containing as much as 10-mg gum resulted in deposits that would place the engines at the point of failure and to the fact that large differences were shown between operation on 1.5- and 10-mg gum fuel, it would seem advisable to keep the ASTM gum content of fuels for continuous use in vehicles as low as possible.

Inspection of the vehicles operating on the two 25-mg gum comparison fuels indicated that the amount of deposits and the degree of engine cleanliness were not affected appreciably by the way the high-gum fuels were made.

Discussion of Illustrations – Fig. 1 presents a picture of one intake valve from each of the Type *A* vehicles operated on 1.5-, 10-, 25-, and 50-mg gum fuel, respectively. The picture illustrates the relative amounts of deposition found on the intake valve tulips at the time of inspection. All these vehicles were operated for 10,000 miles.

Fig. 2 shows the intake port of the valve pictured in Fig. 1 from the vehicle operating on 10-mg gum fuel.

Fig. 3 is a picture of one intake valve from each of the Type *B* vehicles operated on 1.5, 10-, 25- (second test blend), and 25- (first test blend) mg gum fuels, respectively. Comparison of the valves operated on the two 25-mg gum fuels shows no significant difference in the amount of deposits.

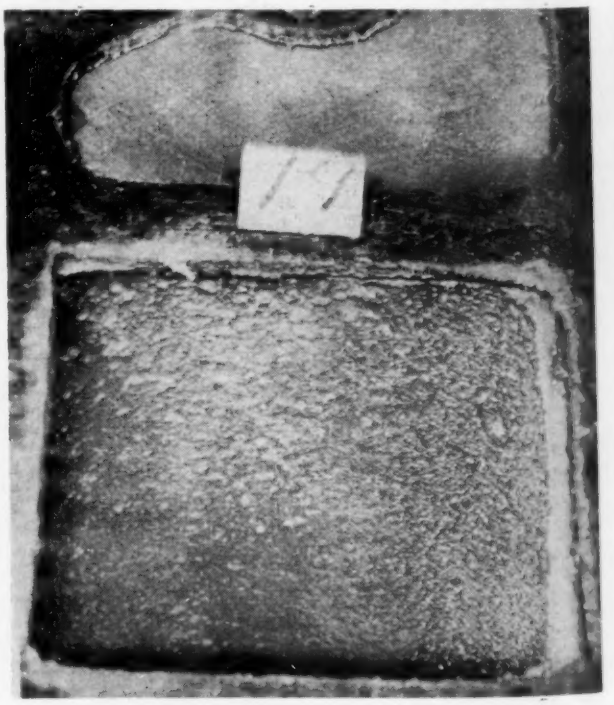
Fig. 4 shows a picture of the intake manifold branch of the Type *B* vehicle operated on 25-mg gum fuel. The branches from the vehicles operated on 10- and 50-mg gum fuels show a similar amount of deposit, but the one from the Type *B* vehicle operated on 1.5-mg gum fuel shows practically no deposition.

Figs. 5 and 6 present a comparison of the deposition in the heat riser sections of the Type *E* vehicles operated on fuels containing 1.5 and 50 mg of gum, respectively. The section from the vehicle operated on 50-mg gum fuel had approximately 20% of the passageway clogged.

Fig. 7 is a picture of one intake valve from each of the Type *G* vehicles operated on 1.5-, 10-, 25-, and 50-mg gum fuels, respectively. There was a slight increase in deposition, but in no case was deposition sufficient to be considered serious. Inspection of other parts showed the same relative differences in degree of deposition with increasing gum content of fuel.

Figs. 8 and 9 are pictures of one representative piston from vehicles *B* and *E*, respectively, operated on fuels of varying gum content. Fig. 8, vehicle *B*, shows a group of pistons which were relatively clean, while Fig. 9, vehicle *E*, presents a group of pistons which were relatively dirty in comparison. It will be noted that in neither case was there any appreciable difference in cleanliness due to the increased gum content of the fuel. Furthermore, although

■ Fig. 4 – Intake manifold (center branch) of vehicle *B* – 25-mg gum fuel



■ Fig. 5 – Heat riser section of vehicle *E* – 1.5-mg gum fuel

Figs. 8 and 9 represent about the extremes in piston condition, this trend did not carry through to general engine cleanliness; that is, with the exception of the induction system, general engine cleanliness was good in both types.

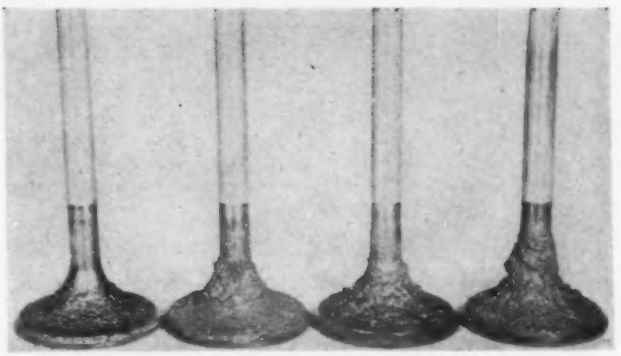
Test 3 – Stationary Engines

In view of the results of the preceding road tests and since it would not be feasible to segregate fuel supplies in the field, it was deemed desirable to obtain data on the use of aged fuels in stationary-type engines, such as those used on generators, battery chargers, and pumping units. Owing to the fact that the road tests had shown fuels containing as much as 10-mg gum caused excessive engine deposits that might affect the operation of the vehicle, it was decided to add a fuel containing 7-mg gum to the list of fuels to be investigated in the stationary engines.

Fuels – The high-gum fuels used in this test contained the same amount of gum as fuels used in the second road test with the addition of the 7-mg gum fuel. The laboratory inspection data on these fuels are shown in Table 3. The high-gum fuels were made with the same high-gum base stock used in the second road test. This base stock had increased in gum content from 30 to 50 mg. Therefore, 17.5 lb of antioxidant per 1000 bbl was added to increase the induction period and to keep the gum content at approximately 50 mg. The high-gum stock was cut back with 2-103B specification fuel to make the 7-, 10-, and 25-mg gum fuels and was used alone as the 50-mg gum fuel. The 2-103B specification fuel contained approximately the same amount of cracked stock as the 2-103B fuel used in the second road test. It was used alone as the 1.5-mg gum fuel. All fuels contained 3.0-ml of tetraethyl lead per gal and 17 lb of antioxidant per 1000 bbl. Lubricating oil meeting 2-104B was used throughout the test.

Engines – Six types of units were used in the test, one unit of each type operating on each of the five test fuels. The various units tested were chosen by the Ordnance Department as representative of their equipment employing stationary engines. They comprised four power units, one battery charger, and one pumping unit. All units were new and were given a run-in of approximately 25 hr of operation on 2-103B specification gasoline. After the run-in period, the engines and fuel systems were inspected to ensure all were clean and free of gummy deposits.

■ Fig. 6 – Heat riser section of vehicle *E* – 50 mg gum fuel



■ Fig. 7 – Intake valves for vehicle *G* showing relative amount of deposits – mg of gum in fuel (reading from left to right): 1.5, 10, 25, 50

Table 3 – Inspection Data

	Third Test				
ASTM Gum Content, mg per 100 ml	1.5	7	10	25	50
Gravity, deg API	59.0	57.7	57.7	57.8	57.5
Distillation, ASTM*					
IBP, F	109	113	113	115	110
10% Point	159	166	163	159	154
50% Point	212	229	235	242	258
90% Point	296	328	342	352	361
EP	378	389	392	397	398
Reid, Vapor Pressure, psi	6.4	5.0	5.4	5.5	5.4
Induction Period, min	1160+	377	202	97	58
Copper Dish Gum, mg per 100 ml	6	52	107	252	421
5-Hr Army Accelerated Gum, mg per 100 ml	30	289	1027	1415	1542
Sulfur, %	0.12	0.13	0.14	0.16	0.21
Octane Number					
ASTM Motor Method	81.5	80.0	79.5	78.5	74.0
CFR Research Method	88.0	86.5	86.0	82.0	79.5

* Per cent evaporated.

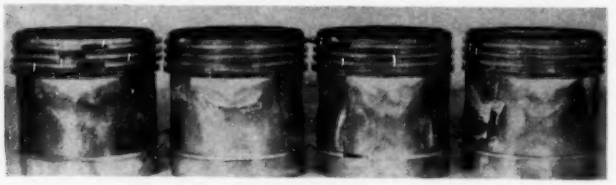
Test Schedule – The test units were operated continuously for 500 hr at recommended load or until failure occurred due to the fuel on which the unit was operating. This constant-speed operation with shutdowns for maintenance at 120-hr intervals was considered to be more severe than the conditions imposed on the wheeled and combat vehicles during the road tests.

Electrical loading of the various units was accomplished by an assembly of water-immersed resistance coils with individual load adjustments. The load on the centrifugal pumping units was obtained by circulating water in a 1000-gal Morang cell for each unit. Test loads were:

Unit	Test Load, kw	% Rated Power Required to Carry Test Load at Start of Test
A	5	100
B	10	40
C	2.5	93
D	1.5	56
E	25	55
F	140 gpm at discharge pressure, 265-270 psi	40

The test loads were those recommended by the Ordnance Department as being typical of field operation. The en-

■ Fig. 8 – Pistons for vehicle *B* showing relative amount of deposits – mg of gum in fuel (reading from left to right): 1.5, 10, 25



■ Fig. 9 – Pistons for vehicle *E* showing relative amount of deposits – mg of gum in fuel (reading from left to right): 1.5, 10, 25, 50

gines were inspected for failure when they were incapable of carrying a load equal to two-thirds of the output set at the start of the test. When a unit failed as a result of the fuel on which it was operating, the engine was disassembled and an inspection of all parts made and recorded. Photographs of significant parts were obtained. Engine temperature data, including oil sump, carburetor air, and coolant temperatures, and the like, and atmospheric data were obtained and recorded. Records were kept of oil and gasoline consumption, power data, and maintenance performed on each unit. Service recommendations and the maintenance schedule set up in the Ordnance Department manuals were followed in so far as possible.

Summary of Results – In general, the results of this test of stationary-type engines substantiated the conclusions drawn from the second road test. That is, fuels containing 7-mg gum or more caused excessive engine deposits and gave every indication that the use of high-gum fuels shortens the duration of satisfactory operation.

The effects of the high-gum fuels were evidenced by heavy, black deposits in the intake manifolds and ports, on intake valve stems above the guides, and the intake valve tulips. Varnish formation on valve stems, pistons, cylinders, valve cover plates, and oil pan sides also was increased by the use of high-gum fuels. As would probably be expected, the various types of engines were affected differently by gasoline gum content. In all the units which were unable to complete 500 hr of operation, failure was due to excessive deposits on intake valve tulips and in the ports and sticking of the intake valves.

The hours of operation of each of the units on the various test fuels are shown diagrammatically in Fig. 10. All units operated on the 2-103B specification fuel containing only 1.5-mg gum finished 500 hr of operation without undue difficulty, with one exception. This unit ran for 415 hr before failure. This type of unit, although not the most susceptible, is critical with regard to high-gum fuels. The earliest failure for any unit of this type occurred at 170 hr with the 25-mg gum fuel, and the longest period of operation was 250 hr on the 7-mg gum fuel.

The unit most susceptible to high-gum fuels had its earliest failure at 80 hr on 50-mg gum fuel and operated for the longest period, 175 hr, on 10-mg gum fuel. Three units of another type operated for 500 hr on fuels containing 1.5-, 10-, and 50-mg gum, but failure occurred at 175 hr on a fuel containing 7-mg gum.

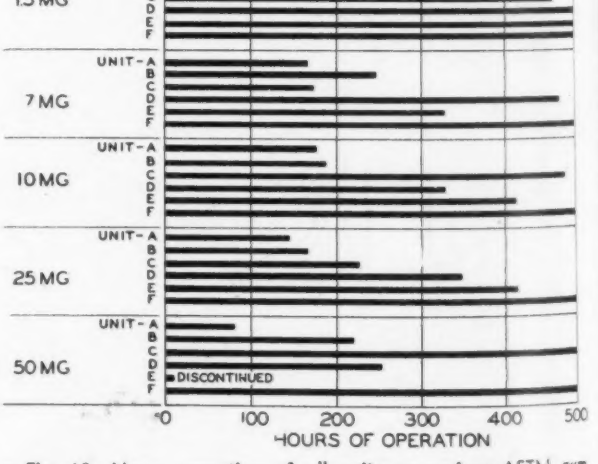
These results further confirm the conclusion that the use of fuels containing 7-mg gum or more places engines at the point of incipient failure due to excessive engine deposits. If it is necessary to use high-gum fuels, there appears to be no significant difference in the effect of fuels containing between 7- and 50-mg gum. As would be expected, in most cases those units with available reserve power ran the longest, since they did not reflect immediately in the load carried, any engine condition leading to the loss of power. However, this was not always true. In one instance, one type of unit operating at approximately 40% rated power output failed in 250 hr or less when operated on high-gum fuels. It should be pointed out that this type of engine in all tests was critical with regard to high-gum fuels.

Discussion of Illustrations – Figs. 11 and 12 show representative effects on intake valve deposits of fuels containing 1.5- and 7-mg gum, respectively. In comparing the figures, it should be noted that the valves from the engine operated on the 1.5-mg gum fuel ran for 500 hr and the valves from the engine operated on 7-mg gum fuel only ran 166 hr.

Figs. 13 and 14 show the intake valves from one type of unit operated for 500 hr on 1.5- and 50-mg gum fuel, respectively. The differences here were not any greater than those shown in Figs. 11 and 12. However, the deposition in the intake manifold branches of these particular units was considerably greater with 50-mg gum fuel than with the others.

■ Conclusion and Acknowledgment

In the overall picture, the detrimental effects of gasoline gum were not in proportion to gum content within the limits used in these tests. Inspection of parts showed a marked difference between the final condition of engines operated on 1.5-mg gum fuel and engines operated on fuel containing 7-mg gum or more. However, very little difference was evidenced in the condition of engines operated on fuels containing between 7- and 50-mg gum. The num-



■ Fig. 10 – Hours operation of all units on various ASTM gum content fuels

ber of hours and miles of operation must be taken into account, of course, when considering the relative appearance of engine parts.

In general, the results of both road and stationary engine tests indicated that, to ensure freedom from excessive engine deposits due to gasoline gum, the gum content of fuels should be kept below 7 mg for continuous use in any type of unit.

The Coordinating Research Council is extremely grateful to the men and the organizations they represent for their assistance in the conduct of these projects.

Particular credit should be given to J. J. Mikita for his assistance in the first road test and to K. Boldt for his assistance and supervision in both road tests. Similarly, the Coordinating Research Council is indebted to W. R. Meese, Office of the Chief of Ordnance, Detroit, for his aid in obtaining equipment and facilities, and to Major H. F. Kley, Office of the Chief of Ordnance, Washington, project officer for these tests.

The representatives participating in the tests were: L. E. Baker, Sinclair Refining Co.; H. C. Baldwin, Continental Oil Co.; J. M. Bigelow, Associated Division, Tide Water Associated Oil Co.; R. A. Bjorndal, California Research Corp.; R. B. Blodget, Associated Division, Tide Water Associated Oil Co.; K. Boldt, Pure Oil Co.; A. V. Cabal, Socony-Vacuum Oil Company, Inc.; M. H. Campbell, Standard Oil Co. (Ohio); W. Fadeley, Office of Chief of Ordnance, Detroit; W. K. Franklin, Shell Oil Co.; J. D. Fuller, Ethyl Corp.; J. H. Fuller, E. I. du Pont de Nemours Co., Inc.; J. Graham, Union Oil Co. of Calif.; J. G. Hall, Pure Oil Co.; S. W. Hauglie, Continental Motors Corp.; J. E. Hickok, Standard Oil Development Co.; I. N. Jones, GMC Truck and Coach Division, General Motors Corp.; Major H. F. Kley, Office of Chief of Ordnance, Washington, D. C.; A. Lenz, Jr., Sinclair Refining Co.; C. R. Lunn, Cadillac Division, General Motors Corp.; W. R. Meese, Office of Chief of Ordnance, Detroit; L. O. Meyer, Phillips Petroleum Co.; J. J. Mikita, Texas Co.; H. B. Minor, Shell Development Co.; G. L. Porter, Union Oil Company of Calif.; R. I. Potter, Standard Oil Co. (Ohio); H. H. Rickles, Standard Oil Development Co.; H. E. Sipple, Shell Development Co.; C. E. Southwick, Homelite Corp.; C. W. Stanford, Hercules Motors Corp.; W. B. Tilden, Texas Co.; G. Trapp, Ethyl Corp.; G. Way, Ethyl Corp.; N. H. Wood, Union Oil Co. of Calif.

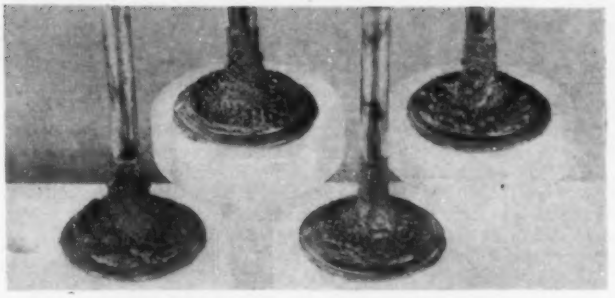
DISCUSSION

Army Said to Experience Little Trouble with Gum

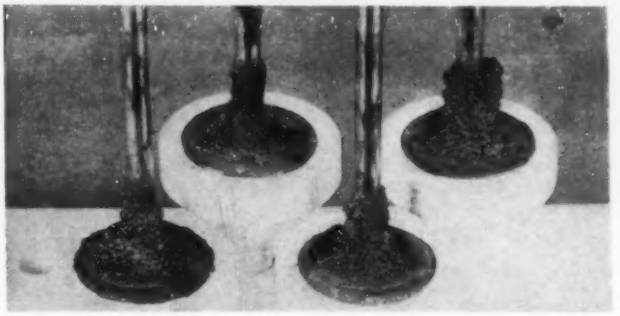
— COM. C. J. LIVINGSTONE, USNR and ERROL J. GAY

Ethyl Corp

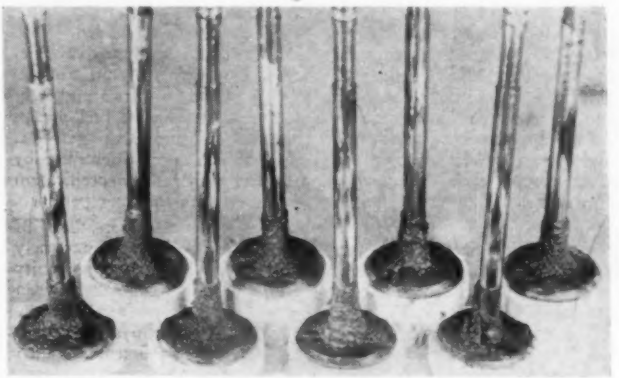
THESE tests were carried out to prove or disprove the emergency recommendations made by the War Advisory Committee of CRC when it was found necessary to store packaged stocks in the theaters longer than anticipated. CRC recommended to Ordnance that gasolines below 10-mg ASTM gum would be satisfactory for any service and that gasolines of 10- to 15-mg ASTM gum might be used for noncombat vehicles, but was not to be used in small stationary equipment. These tests as well as theater experience have verified the wisdom of the CRC recommendation and the Army decision to put the recommendations into effect. It further proves the Army's point that it was not safe to gamble on the possibility of gasoline in combat theaters becoming gummy.



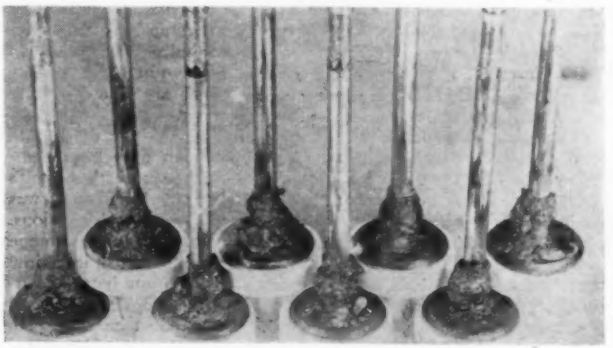
■ Fig. 11 – Intake valves for vehicle *A* – 500 hr of operation with 1.5-mg gum fuel



■ Fig. 12 – Intake valves for vehicle *A* – 166 hr of operation with 7-mg gum fuel



■ Fig. 13 – Intake valves for vehicle *F* – 500 hr of operation with 1.5-mg gum fuel



■ Fig. 14 – Intake valves for vehicle *F* – 500 hr of operation with 50 mg gum fuel

In the summer of 1943, considerable objection from suppliers was raised to a modification of the 2-103B specification which would then call for 7-mg ASTM gum and a 420-min induction. We are sure that, rather than any intentional disregard of the need for safe operation of Army equipment, their objections were based (a) on a lack of understanding of the Army problem in the field and (b) on the difficulty of making such fuels available in large quantities in view of the aviation gasoline program.

Later, the 2-103B specification was modified to read 4-mg ASTM gum and 480-min induction period. This was done because of a decision to prohibit the shipment of gasoline from U.S.A. ports to overseas areas which contained more than 7-mg ASTM gum. Thus gasoline could be purchased at 4 mg and stored for some time in the U.S.A. before the gum would rise to 7 mg.

The emergency recommendations of CRC were very useful to the North African theater where gasoline stocks were held in storage longer than anticipated because of failure of the enemy to retreat as rapidly as the Allies would have liked. Due to a joint British-American operation in the Mediterranean Theater and the use of some British equipped gasoline testing laboratories, the British IPT gum test procedure was used. This test is somewhat more severe, so that it is generally accepted that the ASTM gum would be one-third less. For example, 10 mg IPT would be approximately 7 mg ASTM. Since the Mediterranean area followed CRC-Ordnance recommendations for 10 mg but used the IPT, it meant that gasoline used in Army equipment rarely exceeded 7 mg ASTM. From the data reported by Mr. Alspaugh, it is indicated that this additional safety factor was worth while. The Petroleum Section AFHQ should be complimented on the way it controlled a serious situation which, if it had got out of hand, would undoubtedly have been reflected in serious casualties in men and equipment during the landings in the south of France.

Personal experience in combat theaters has shown the soundness of the above recommendations. Unfortunately, none of the tests described here were up rated at the high average ambient temperatures which were experienced in North Africa and Italy during the summer, or the interior of such countries as India or Burma. The average temperatures in the United Kingdom, France, Belgium, and Germany areas were such that storage of gasoline and the formation of gum were not problems. There was the exception of a small amount of gasoline shipped into the United Kingdom before the adoption of the first 2-103B specification calling for 7-mg ASTM gum and the 420-min induction period. Also, certain vehicles arrived in England with high-gum gasoline in the storage tanks. However, several million gallons of gasoline was stored in North Africa, some of which formed excessive gum. This gasoline was blended off in accordance with CRC-Ordnance recommendations and no difficulties were encountered. There are on record, however, the gum troubles of a large number of vehicles which were stored in Africa and later shipped to Italy. These vehicles had gum troubles with fuel pumps and carburetor needle seats due to gum accumulation in the gasoline. One point that has not been fully explored in the tests described by Mr. Alspaugh is the storage problem when higher than specification gum content gasoline is placed in stored vehicles for long periods.

Not too much trouble from gum was experienced in the CBI theaters due to the fact that the turn-over of gasoline was extremely rapid. However, on pipe line engines we did have an opportunity to observe the action of gasoline varying in gum content between 4 mg and 6 mg ASTM. These engines, operating for several months out of the year with carburetor air intake temperatures at 125 to 140 F, would accumulate sufficient gum deposits in the center branch of the inlet manifold to reduce power output so that pumping loads could not be maintained. From the description of these present tests, it seems apparent that on wheeled vehicles a large percentage of the gum comes out on the inlet manifold hot spot, provided the temperature at this point is fairly high. Otherwise, a considerable portion of the gum comes out in the inlet port and on the under side of the inlet valve. There have also been indications from field observations that gummy gasoline or fuel of low inherent stability may cause light deposits on exhaust valves which are serious because such deposits induce valve sticking and burning.

The author does not need to make any apology for having to blend gasolines of both high and low gum content with fresh gasoline to run these tests. Such a procedure is in line with the Army's methods of handling gasoline, since gasoline of high gum was blended back to the CRC limits in the theater on several occasions. It should be called to everyone's attention that the severe storage stability requirements of the 2-103B specification resulted in a low proportion of components of high potential gum in any batch. Handling, in bulk and shore tankage, mixed gasolines from several sources so that the inherent stability of the product reaching the theaters was extraordinarily high. Consequently, one would not expect the rate of gum formation after inhibitor depletion in Army gasoline to be as rapid as it might be in some civilian fuels made totally from components of high potential gum formation. It should also be pointed out that gum from the latter might have caused more engine difficulties if included in these tests. It is our belief that gasoline in which it is difficult to control gum formation may contribute to serious engine deposits even though the ordinary physical inspection at the time of use might not give such an indication. In other words, a gasoline of low inherent stability may show its lack of stability either by formation of dissolved gum in the gasoline in storage or by chemical change leading to engine deposits while passing through the engine. In either case engine efficiency, and thereby the mobility of the military forces, would have been lowered.

Peacetime civilian operation can obtain an important lesson from these tests conducted by CRC for the Army. No truck or bus operator is satisfied with 10,000 miles before the need of engine repairs. Mileage on engines varies from 25,000 miles to close to 100,000 miles, depending on the type of equipment. The gum content of gasoline is an important item to the fleet owner, and the difference between 1.5 mg and 7 mg can make a great deal of difference in his operating cost. We believe industry, both automotive and petroleum, should be grateful to the U. S. Army Ordnance and CRC for making this up-to-date and extensive work available. However, since these tests, as pointed out above, did not include high concentrations of potential gum-forming components in the fuel tests, it is not safe to assume that 5000 miles of operation can be obtained with any civilian gasoline in which the gum content ranges between 7 and 50 mg. It is our belief that gum is like a disease; it can kill slowly or in a few days, depending upon the circumstances of gasoline composition, operating procedure, and temperature conditions. In any case, the use of fuels of high stability is recommended.

Test Results Led To Fuel Savings

— CAPT. NORMAN L. KLEIN
Office, Chief of Ordnance

THE CRC has assisted the Ordnance Department with many urgent problems during the war, among which gasoline stability was of prime importance.

The successful determination of the gum tolerance of Ordnance vehicles, the story of which has been presented by Mr. Alspaugh, permitted the maximum use of depreciated stocks overseas and prevented operational difficulties that might well have resulted in serious consequences under combat conditions.

COMBUSTION STUDIES

continued from page 288

equipment all affect the smoke meter readings. All values presented in this paper were made at one engine speed and the number of variables reduced to a minimum.

Smoke emerging from the exhaust may be referred to as the smoke point. A diesel engine operating at high values of bmep (compared to maximum output) produces black smoke. By reducing the bmep, the smoke points stays within the cylinder. However, if the exhaust valve opened earlier smoke would emerge. There is little doubt that large quantities of carbon are present within the combustion chamber during the combustion process, as evidenced by the rich flame color and absorption measurements through the combustion chamber. If this carbon is not entirely consumed it will emerge with the exhaust. Since the evaporation is only a fraction of the total ignition delay, fuel vapor pockets exist which are subjected to the high temperature. It therefore seems probable that chemical reactions take place in the fuel vapor in addition to the normal reaction with oxygen. At the elevated temperatures during combustion, fuel vapor breaks down rapidly with carbon as one end product. It should also be brought out that fuel deposited on the cylinder walls or the piston will evaporate slowly, causing late burning and hence, smoke. This will also produce fuel in the exhaust, depending on load and ambient engine temperatures. Engines operating at idle or at low bmep show that fuel comes out with the exhaust. Of course, it must be recognized that the volatility of the fuel influences the evaporation rate. There are also tarry products present in the exhaust which exhibit polymerization reactions of the fuel. A fuller explanation of the fuel reactions during the high-temperature condition of the combustion chamber requires more experimentation. When Mr. Schweitzer states that a rich mixture causes smoke, he agrees that a high concentration of fuel relative to the oxygen content is one cause of smoke. This smoke appears as carbon which is an indication of fuel decomposition.

One experiment with cetane and blends of cetane with alpha methyl naphthalene and cetane with benzine brings out an important point in regard to the discussion. Fig. A shows the smoke, as represented in Part I of this paper, as a function of bmep. These three fuels presented almost identical smoke patterns. Consequently, reducing the cetane number by blending cetane with either alpha methyl naphthalene or benzene changes the smoke very little. Additional experiments along this line woul certainly add more information on smoke.

MOTOR TRUCKS of the FUTURE

by ROBERT CASS
Assistant to the President
White Motor Co.

I AM a little conscious of the degree of prognostication accredited to me by the title because, although it is very pleasant to have a free range with respect to what may happen in the future, nevertheless, I have many times during the writing of this paper found myself wondering just how far into the future we should practically consider. Past experience seems to indicate that at least five years hence is a reasonable period to consider, and I have endeavored to provide some food for thought with what is hoped is the proper degree of attention to the economics involved. The subject matter of this paper does not represent my company's position but the ideas are of a number of engineers in various sections of the truck field.

The prime purpose of this paper can only be that of directing thought toward possible lines of endeavor which will come up for review in the truck development picture from now and during the period stated. Certainly, we need to take a good hard look at the economics, as some of our accepted standards will undoubtedly be changed, and will have to be changed, if the development is to be along some of the lines indicated.

We shall also have to take a good hard look at what is apt to develop in our highway system, as the highways naturally will greatly influence some of the possibilities of the changes contemplated. The economics are also bound in the highway development – a simple illustration being that, if today, under existing rate structures, a common carrier finds it unprofitable to carry a certain kind of freight more than a distance of, for instance, 850 miles, he may find, with the superhighway development contemplated, this distance can be greatly extended.

You have already heard or are due to hear the story of the work that has been done in the past by the Public Roads Administration of the Federal Works Agency which, in addition to setting up a program of mileage in the greatest possible detail, has also established standards for the superhighways which can so readily change the truck transportation picture of the future. It is safe to say that if roads are built, and unquestionably they will be, within the limits of reasonable standards set up by the Public Roads Administration, speeds of even as high as 70 mph would have to be contemplated in the design of the various component parts of our motor trucks. Again this is presupposing that such highways, by virtue of their construction, would be outside the usual speed limits that operate on ordinary roads, and also it should be noted that, in this consideration of superhighways and the speeds involved, every precaution has been taken with regard to safety and reasonable construction costs so that the premium for high-speed operation should be neither at the expense of safety nor in high road costs.

Perhaps before getting into the main subject matter of this paper, a little background on the industry itself may be useful in establishing the scope of the operation affected by these future developments.

We are talking of a transportation industry that involves the use of around five million vehicles and which, from a manufacturing, maintenance, and operational standpoint, involves at least six million men, which, with their families, represent a good sized segment of our total population. The investment each year may well run into a billion dollars in new vehicles, based on an annual replacement of around three-quarters of a million trucks. It is easy to see how any change in the economic picture of this industry from the standpoint of design and manufacture can and does have a very large effect on the cost of living, as distribution costs are a major item in commodity prices.

It would also seem reasonable to assume that we should consider truck development, by and large, as a separate development from that of the passenger car. Our engineering efforts should, therefore, be directed to considering all the component parts as only truck items. We should,

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 8, 1946.]

THERE is a great deal of room for improvement in future motor trucks, Mr. Cass reports, both along already established lines and in the direction of specialized models for specific purposes.

Possible individual improvements actually omit no aspect of the truck from powerplant to driver comfort.

Whether or not the gas turbine will have a place in future trucks depends, the author believes, on how it stands up against the future reciprocating engine with regard to the considerations of size, thermal efficiency, maintenance, weight, and initial cost.

THE AUTHOR: ROBERT CASS (M '39) is now assistant to the president of White Motor Co. A native of England, where he received his engineering education, he came to America in 1924 to join the faculty of Harvard University as an engineering instructor. The following year he joined the White company, and has been successively assistant chief engineer and chief engineer.

therefore, think of the development of the vehicles of the future in terms of single-purpose tools in which the trucks for a particular industry which depends on transportation would be designed definitely and solely for that industry. To some degree, this has already been started by the designs for door-to-door delivery and it may be equally important to design a vehicle for over-the-road operation which will bear no resemblance to our present designs and present methods of manufacture. In any case, anything that is presented in this paper will be against the background of the economic necessity for such consideration, and in every case, as an overall picture, it will attempt to see how this can be effected. We, therefore, start out with the truck industry in which, as a major industry, any changes made can have as far-reaching an effect as those changes themselves. A typical case could be a chassis made wholly from light-weight metals.

■ Engines

With regard to the development of the units, it would seem that we should first consider what changes, evolutionary or revolutionary as the case may be, may take place in the powerplant.

There has been much discussion of the very startling developments during the war of the gas turbine as distinct from the turbo jet, and many claims have been made in general with regard to its possibilities as an engine to be adapted for commercial use in ground transportation in the future. Certainly, if it can show an improvement in the five major considerations in engine design, it has to be considered as a future possibility as a power-producing unit first, without relation to how this power can be utilized or applied.

Five basic considerations would seem to be: first, size; second, thermal efficiency as represented by the miles per gallon; third, maintenance of engines; fourth, weight; and fifth, first cost. First cost is placed last because if any marked improvement could be shown in the first four as compared to what we can expect from the further evolution of the reciprocating engine, obviously first cost is purely relative.

Consultation with people interested in the fundamental development of small-size gas turbines seems to indicate that there are no major drawbacks to making the powerplant in sizes perhaps of 300 hp occupying a smaller space than now occupied by a similar reciprocating engine together with its cooling system, and so on.

At the onset, it is considered reasonable that we could stand a higher specific fuel consumption, as the fuel picture lends itself to the use of lower price fuels, both oil and gasoline. The specific fuel consumption of the gas turbine at the present time is roughly twice that of our orthodox truck engine, and even though the fuel used may be less in cost per gallon, real progress can only be measured in an increase in output per cubic inch for each pound of fuel used.

In the sizes contemplated for the truck picture, they see no need for a cooling system, as the temperature suggested – 2000 F – would be maximum and any added efficiency which came from cooling and regeneration of any kind would add relatively little to the total overall economy picture but would add a lot of complications.

It is anticipated also that the cost picture will be favorable once the regular automotive methods of production are applied. At the onset, it is anticipated that the pressure ratio could be five to one and compressors available in the size necessary for this are relatively inexpensive. In the turbine itself also, the question of cost seems to be reasonable in the horsepower range we are considering as maximum at this time. We might have difficulty in connection with the cost in smaller sizes, but in the future program this could be merely a question of production tooling. Certainly, from the design standpoint as affecting maintenance, the overall picture is one of great simplicity, and those who are enthusiastic naturally feel that none of the other problems are too great. Lubrication is thought to be a very simple matter as compared to our present engines and the tolerances in manufacture are in the neighborhood of 0.010 or 0.020 on blades, and such parts, rather than the extreme limits of accuracy found necessary in the power-producing parts in reciprocating engines.

One of the principal difficulties as far as the engine itself is concerned, revolves around its speed, which obviously must be very much greater than that of the reciprocating engines – at least in the ratio of four or five to one.

Whilst it is possible to mount the rotating shaft on antifriction bearings and it would be desirable to do so, the question of life of that part constitutes a problem, as the peripheral speed of the balls is very high and the life at the present time in aircraft installations is relatively low in terms of hours. Figuring on what should be a normal life expectancy of such bearings, it seems that we must expect a life of 5000 hr as a minimum. This may be regarded as very high for a bearing running at perhaps as high as 20,000 rpm when the diameter of the shaft will be in the neighborhood of at least 3 in., but the manufacturers of such bearings are not too worried about this as lubrication is good.

The characteristics of the engine, however, call for a maximum speed variation from the top speed of no more than 25% if anything like efficiency is expected. The question, therefore, of power transmission in ground transportation becomes one of electric or some other form of torque conversion not yet developed.

Whilst the gas turbine can be considered as a possible engine for the future in trucks of large horsepower, it awaits equally fundamental improvement in the transmission units. It is possible to run polyphase electric generators at very high speed when the efficiency is relatively high, yet, the electrical problems would seem to be quite definite and at the present time offering less possibility of solution than those that go with the gas turbine itself.

The size of the unit for 300 to 400 hp, it is believed, could be in the neighborhood of 36 in. long and its diameters occupying much less space than now taken up by our reciprocating engine. It is desirable, but not necessary, to have the burner on the same axial plane between the compressor and the turbine, but obviously it cannot be too remote if the highest efficiency is to be maintained.

We have a still further problem in connection with the gas turbine, regardless of type, which involves metallurgy, and whilst significant changes will have to be made in the metallurgical quality to secure any future engine development contemplated, the problems are much greater with the turbine than, for instance, with the valves in the conventional type of engine.

It must be remembered that the standards already established in the motor truck field are high – 100,000 miles being such a common figure before any major work is done on an engine that it has become the minimum con-

sideration of the engine designers today, and we are constantly looking for changes that will increase this mileage. The gas turbine, therefore, would seem to be facing these very serious problems of development. On the other hand, there can be no gainsaying the many advantages that could come from such an essentially simple engine.

With respect to the development of the conventional engine for truck purposes, there would seem to be enough reason to expect such changes in fuels in the years stated that, with supercharging, will make possible running the bmep up at least to 180; and with this large increase in engine output for truck operation, it is expected that maintenance standards now existing can be held and perhaps improved.

Certainly the economics of the truck transportation picture demand that in some way the power transmission line be reduced in weight and cost, and the price per horsepower of the engines is expected to be reduced, as it has been constantly over the past years. It is expected, therefore, that engines of around 600 cu in., now used in trucks, can be reduced in size, using fuel and supercharging developments, to something over 400 cu in., and in so doing bring about not only the reduction in price per horsepower but a reduction in the cost and weight of the units behind the engine. Of course, this will call for a consideration of improvements in a number of very vital parts in the engine that would make this program possible or impossible.

With respect to the range of engine power that the future trucks will have, 300-hp truck engines are not by any means an impossibility. In fact, engines of that horsepower are now being used somewhat experimentally on the Coast. Weight per horsepower, using common materials, of between five and six pounds in future engines is very desirable. To make this possible, we shall, of course, require a fuel of 90-95 octane and lubricating oils of such a quality as to handle temperatures well over 250 F. It is recognized that the high bmep's which have been mentioned may occasion, in their early stages of development, some roughness, but there is every indication that with the combustion-chamber shapes as well as the introduction of stabilizing compounds through such devices as the vitameter, which was discussed in a paper given before the SAE last year, this roughness may be readily controlled.

There are some specific parts of engines which should be reviewed at this time:

Piston Rings – Piston rings, which are so vital in future designs, are under the spotlight. Single-cylinder work indicates that the life of the rings is still far short of what we need in our future high-output engines, assuming that they must be reasonable in price as compared to present-day prices. It also seems likely that temperature conditions will be more of a factor than the pressures involved. Whilst chrome-plated rings seem to be a step in the right direction, there are, in the opinion of the ring manufacturers, other lines of development which seem more profitable to pursue.

Compatibility of materials has a bearing of considerable importance on this subject, and there, cylinder block or liner material, as well as finish, also becomes very important. A reasonably priced ring to withstand high temperatures and pressures and still have as good a life as found in present-day engines seems to be a distinct possibility.

The ring people also feel that the greatest aid which can be given to the functioning of piston rings is to develop cylinders and pistons which will remain as nearly symmetrical as possible to their original shape under operating temperatures and to provide cooling that will keep the temperatures as low as possible. Oil cooling of pistons suggests one way of controlling high temperatures.

Pistons – The piston picture is similarly felt to be one that also demands a great deal of attention. Again, in the research picture, standard design pistons satisfactory in present engines fail, at the higher bmep's and temperatures, to give life comparable to standard expectancy, and experience indicates that new materials and new methods of manufacture should be considered. Again, we cannot consider aircraft prices or aircraft-engine life as criteria.

The clearance problem of the piston is also one of concern, and some positive control features should be built into them so that minimum clearances prevail.

Extruded or forging perhaps offers a solution, although, as stated in the beginning, this paper deals with the requirements or problems and is not concerned with the manufacture so long as there seems to be a good chance of fully meeting the issue. Specifically, in the case of pistons for truck engines being considered, the piston manufacturer feels very confident of meeting the requirements.

Bearings – With respect to the high-output engine we are considering, one of the vital developments which must come, of course, is that of a change in bearing construction or materials used in main, connecting-rod, and wristpin bearings. Of these, the one which will be the crux of the development is the crankpin, and it seems certain that the straight copper-lead bearings that we now use will have to give way to another alloy or alloys. It would seem desirable that we should consider at least the possibility of a 50% increase in bearing load, with an overload running as high as 75%. Discussion with various sources reveals a great deal of research being done along the line of producing bearings, such as are used in aircraft engines, which are capable of standing intermittent loads with pressures as high as 7000 psi. Their present idea is to develop a combination perhaps using such alloys as are used in the aircraft bearings with such disposition and process of manufacture as to call for a price increase over our present truck engines, but certainly not more than twice what is now paid for good copper-lead. It is obvious that if that is the thinking at the present time, when the use of these bearings becomes a matter of fact, one will see the price differential between the two bearings become very much less. We, therefore, then have possibilities of new alloys for bearings, including the trimetal bearing, and perhaps the introduction more generally of aluminum-alloy bearings disposed on such backing as to remove some of the objections of the present type of aluminum bearing.

It is interesting and comforting to note that we can cut down the width of the bearings considerably without any sacrifice of life, assuming the bearing materials keep up with the pressure increase. Lubrication problems are easier with the narrower bearings, although control of oil both as to quality and amount becomes more critical.

Actually, as far as given materials used in bearings in the past are concerned, the safe loadings have been decreased because of the longer sustained high-output conditions of the present truck operations. This will be still further extended on our future high-speed highways. The Pennsylvania Turnpike has confirmed this, and many life expectancy standards have changed as a result. Again, however, the engine designer, in his attempts to boost the

output per cubic inch, is still on good ground as far as the vital engine parts are concerned.

Valves – To make possible the output desired, we are obviously going to depend on the valve picture improving as far as kind and cost are concerned. Liquid-cooled valves are not new to truck engines, but our future unit will require more and more such materials as sodium, and we must still get them at no greater cost than now. In fact, we should expect them to be lower in cost.

We have long known the effect a cool exhaust valve can have on the octane requirement of an engine, and with the future engines becoming more critical, both inlet and exhaust valves are due to be considered seriously in the light of kind and cost.

The viewpoint of the valve manufacturers is an encouraging one. They, however, still have the problem of dealing with compounded fuels as far as valve seat deposits are concerned and the development of a stable rotating mechanism is one effective way of dealing with this. Indications in this respect are very favorable and with the need for this possibly rapidly increasing, several solutions of the problem may be expected.

Supercharging – Without attempting to discuss the relative merits of types of superchargers and their driving mechanism, there are some indications that the most desirable is the centrifugal type. This is because it can first be more readily driven from the engine exhaust system and we can thereby get the boost at the time we want it.

Fundamentally, the supercharger installation should not cost more per horsepower gained than the cost of the same power increase in a larger engine. The weight-saving possibilities of the supercharger should be considered a plus.

Injection System – This is obviously a part of future engine development which should be pursued with vigor in the next five years both as to cost, which at present is much too high, and to the characteristics which will have a marked effect on maintenance. A carburetor is a simple instrument to maintain. Further, it would seem that with a gasoline engine designed for an injection system, size for size with a carbureted engine, we should obtain at least a 25% decrease in fuel burnt.

Diesel – Up to now we have mentioned only gasoline engines, and because it seems that the major development in reciprocating engines, not forgetting the two-stroke, will center around that type of engine, the diesel has not been included. The fuel picture is predominantly in favor of striking gasoline engine progress in truck engines and there are other economics outside of fuel that further influence our thinking. Normal refinements can, of course, be expected in diesel engines.

Cooling – One of the developments which calls for serious consideration is that of cooling – both of the engine and the oil. It would seem desirable to increase the operating temperature of our engines, and this development will lead to the use of other solutions than water. Given proper plumbing and pumps designed for pressure operations, the possible reduction in size and cost of even the conventional type of radiators is intriguing. We need an oil cooler badly for future engines – a simple, easily cleaned unit that takes up no more room than a generator does now. The chemists in the oil industry should get knee-deep into both these problems, as a good solution of both depends largely on them.

The transmission problem is a very complex one, in that from the inception of the straight spur-gear transmission with the simple gear shift lever, considerable attempt had been made either to refine or obsolete that form by various forms of converters, automatic shifting devices, and various other innovations. However, the truck transmission as of today is basically not a great deal different than from any years in the past.

There is, of course, naturally at the present time, a great deal of speculation in the truck field as to the use of automatic transmissions. This comes from two sources – one being due to its development in the passenger-car field and the other in the bus field.

Looking into the future development picture, it would seem that we should replace the present gear boxes ultimately with some easier form of handling gear changing, particularly in city work and certainly in terrain where a great amount of use is made of the transmission ratios. The possible economies of a properly designed automatic transmission bring it definitely in the future picture. However, it would also seem that for this to be effective in the sense just referred to, other than speed must be the means of changing the ratio, because the ideal situation for a transmission would be that of changing taking place quickly and accurately depending only on the power requirements and where as much as full load would be maintained on the engine in any gear so as to assure the maximum economy possible.

To do this, however, up to date, involves complication way out of the average truck picture, and at the present time maintenance could vary from a few automatic boxes in a small fleet to a larger number in bigger fleets; and it is obvious that there cannot be enough specialists of the kind at present considered necessary with the transmissions now being currently used in the bus field, and neither could the advantages show savings enough to justify cost and upkeep. The bus picture is a special one because of the maintenance setup and the paramount need for it. It is, however, suggested that in the period covered by this paper, very serious consideration be given to the automatic transmission, and if possible using a fluid means of transmission. This item of the transmission has been demonstrated as having practically no special maintenance problems and, therefore, it makes its inclusion in an automatic transmission almost a necessity.

Whilst, however, at the moment the cost picture looms large, the history of this item is similar to others, in that ways and means will be found to handle that phase. The synchronizing picture is also one of development for trucks in spite of the bus development.

A well-known transmission maker says: "In the next few years I believe you will see the means of shifting improved by synchronizing it or a means equivalent, and power shifting worked out. I think it might be well to point out that the synchronizing of truck transmissions presents some problems that differ quite a bit from the same approach that we made to other major improvements. In other words, if you take the passenger car years ago, it followed the same steps that later the truck and bus transmission went to; that is, first, spur gears; the next major change, helical gears; the next, synchronizing; and now, the automatic power shift, and so on.

"The truck transmission is now catching up to the last two phases of the passenger car, that is, the synchronizing and improved methods of shifting. It was relatively easy to put in helical gears in truck transmissions, and without too much developing, they did a pretty good job. It was

not too difficult to put in synchronizing units in bus jobs, but when we applied the synchronizing units in a truck it was quite often the opinion, 'Why is it not easier in a truck than in a bus?' A careful analysis will show that the ratios between steps on bus jobs are much closer than a truck job. For instance, a four-speed bus job with a low reduction of 4 to 4½ is about as far as you have to go, whereas, if you took a four-speed truck job, you would start out with at least a 6½ low, and if it is the case of a five speed, you would start out with at least an 8 to 1 low. This means that in the process of shifting, you have to synchronize greater steps between speeds and absorb these kinetic energies through just as short a span of time as is possible. Therefore, in proportion, when you think in synchronizing truck jobs, you have to realize that you have some inherent problems that you cannot base previous practice on. Your ability factors to move the loads in trucks are much lower than with passenger-car practice or bus, and because we loose the inertias so fast at low speeds with gross loads, we have to shift relatively fast. This is not an insurmountable problem, but will have to be met by careful study if we are to get not only synchronizing value but also get any life in those synchronizers.

"When we touch on the life picture of a synchronizer, we must realize that we go out of our way to design a piece of mechanical goods to break oil films fast and get high pressures on cones, or the methods we use to create enough friction to cause two members revolving at different speeds basically to stick together. This is the last thing that we would do if we were designing a bearing to get long life and service. In the next few years I think this will be worked out, and while at first the cost is going to be there, its true place in industry will not be the cost but what its economic value is.

"The power shift problem is going to be worked out, I believe, in a manner that we can control that power and achieve that, I believe, relatively and economically. I think it will be a long time before the full automatic or the torque converter with automatic gearing will be developed for the rank-and-file load job. In off-the-highway vehicles, the torque converter will no doubt have its place, but further development must be given to it before it is even economical for that type of truck."

■ Suspensions

Earlier, we predicted that the motor trucks of the future should be safe to operate at speeds as high as 70-80 mph, and this involves a detailed study of suspensions, steering gears, tires, and braking.

With regard to suspensions, there would seem to be a need for taking a good long look at the possibilities of changes in this respect. For years, in the truck field, we have tried to improve, as much as possible, the design of the common leaf springs with an idea that such design features should promote safety and under the worst operating conditions, keep the stress down to reasonable limits. In general, little has been done in designing a chassis that will enable us to get good operating conditions with both light and loaded vehicles. The application of shock absorbers to existing leaf springs would not seem to be an answer to the vehicle of the future, particularly when operating at high speeds when the vehicle will be vulnerable to road shocks to a greater degree than we now consider usual. Driver fatigue plays a very important part in the consideration of suspensions as well as the hazards of side sway, the shifting of load centers, and certainly we need more effective control over the safe handling of perishable and fragile commodities. The problem of absorbing shock without disturbing the sprung mass is not a simple problem, but actually complex. Certainly it is of the utmost importance that axle and steering geometry be considered hand in hand with the suspension rate problem, as it is closely linked with axle geometry and steering stability. Roadability, which after all means safety, is dependent upon the successful development of these various functions in a simple, foolproof, dependable structure.

There are certain fundamentals that are well known but should possibly be restated. Spring rates, in whatever form used, must be lowered to improve the ride under light load conditions and at the same time the spring must provide a progressively increasing rate as the heavier loads are carried. The problem of preventing a wide change in platform loading height from minimum to maximum load must also be considered. In relation to the overall ride conditions desired, it is suggested that hydraulics be developed to provide an automatically compensated chassis height regardless of load, and whilst such devices have been proposed in conjunction with the air spring, its complications have been too great to be seriously considered. We must also remember that a vehicle that has a low spring rate and in which the ride correspondingly seems better, in the past, has always seemed to achieve this improvement at the expense of the stability of the vehicle. We shall probably make progress along the lines in the future of relieving the springs of the job of starting and braking torque and concentrate on having them do their one job of absorbing the road shocks. Controlling side sway by automatic control is a very definite possibility in trucks of the future.

As we consider the types of springs available – leaf, torsion, coil, rubber, air, and volute – we certainly cannot complain of lack of variety as far as design possibilities are concerned.

In connection with the use of rubber for suspensions, forward-thinking men in the rubber industry are definitely of the opinion that much work lies ahead of them but it is very promising in its possibilities. Rubber used in suspensions as an elastic element can be used in many ways: in direct compression, rotationally compressed elements, torsional shear in various forms, direct shear in various forms, and combination of shear and compression. We also look for development perhaps of the bellow-type spring which lends itself to innumerable treatments in the interests of choice of rate, increasing rate, and choice of frequency. The further attraction of rubber can well be that it can be employed at points of attachment eliminating lubrication and noise.

■ Tires

Rubber seems to be standing at the threshold of great developments, and this leads us to a consideration of the tires. It has long been known that speed naturally is a destructive element as far as tire life is concerned in that there is an increasing tendency for the tires to heat up to the danger point. Yet, in spite of this, the curve of actual life using thousands of tires as a basis, indicates that, with the increase of speed, the tire life has always, with but one exception during the last 10 years, gone up considerably,

the gain never falling much lower than 5-6% and sometimes being as high as 10%. Looking into the possibilities of tires for these vehicles we are considering, we see the possible use of different rayons, nylon, and steel wire. Wire is 10 times as strong as cotton which, for a long time, was the standard tire cord. We are all well aware of what has happened to rayon tires, and we may well look forward to other types of synthetic cords which have a promise of even more attractive results. The steel wire, which would have a greater appeal to the mechanically minded automotive engineer, is already in a well-advanced state of development, and in the period of years we are considering, its development unquestionably will offer us safer tires with longer life even at these increased speeds.

The war has also shown us how devices can be used to vary our pressures, and unquestionably work will be done along these lines to simplify it. One day we may be able to have automatic control of pressure by the driver and, therefore, temperature control whilst under actual operating conditions. The safety aspects of this are certainly of tremendous value. With regard to heat, the enemy of rubber and its cords, we may find ourselves, in this future period, using light-weight metal rims which will have the advantages of better heat conductivity as well as obvious reduction in weight.

■ Steering

With regard to steering, it seems reasonable to assume that we are reaching the end of the straight mechanical gear and have reached the point where there is a necessity for quicker and easier control in future vehicles. We may be wise to consider the development right now of suitable power steering gears in which weight and cost come more nearly within the scope of the rest of the units used in these future trucks. I think it would be only fair to say, at the present time, that power steering is getting more attention than it deserves as its use is now very limited but the spectacular results may call for power steering development to go ahead at a much higher rate. Our present steering gears, generally speaking, are very good, but it is the general consensus of opinion that even as good as they are, steering, in general, demands too much physical effort and some power assistance must be considered in the future. This can take the form of hydraulic or electric power and it also can be said that there is, within the range of practical possibilities, during the next year or two, the possible development of a mechanical steering gear which offers promise. This device referred to operates on an entirely new principle and, if successful, may easily serve to replace power steering units, particularly on medium- and heavy-duty trucks in highway service. There is no reason why it should not be used on off-the-road service also. It is interesting to note, however, that we have reached the stage where we can confidently look forward to a great deal of development from the angle of safety regardless of speed in our steering mechanisms.

■ Braking

With respect to the question of braking for these large vehicles running at high speeds, it would seem highly important that the developments of brakes of an entirely different form than we now use be considered as the ability usefully to dissipate and handle the large number of Btu's required to stop a train carrying 70,000 lb from 70 mph is of such a magnitude as to make us wonder how stable present-day brakes would be and how long a life they would have. Any brake designs today, as far as we know, must follow the pattern of rather significant improvements of the general principles of braking, but at the same time the fundamental problems of brake drums – heat and checking, and so on – are still with us so long as we have the friction type of brakes. The hydraulic and electrical fields may well be looked into in this respect and it will certainly be necessary to have a thorough and intensive examination of fresh materials if the existing drum type of brake proves to be the only solution from the point of view of cost, weight, and maintenance, all of which must be taken into consideration.

■ Driver

Undoubtedly, much of the success of our truck transportation picture of the future will be in the hands of drivers. We should, therefore, first consider some of the things that should be done in the years ahead of us to promote safety and comfort and ease of control to the driver. This will call for a complete reorientation of our thoughts with regard to the size and shape of the driver's compartment because, not only are our present jobs when used at high speeds definitely expensive to operate from the standpoint of the wind resistance and, therefore, occasioning a waste of horsepower, but also they are so constructed as to have relatively little comfort for the over-the-road service. We shall, therefore, envision a compartment for the driver in which as much study has been given as that of the compartment of an airplane pilot.

There is slowly amassing a great deal of data with regard to human fatigue under various conditions, and all of these data will undoubtedly concentrate itself into providing: first, a fully adjustable comfortable seat; second, a clear vision on the sides as well as the front; and third, an air conditioning system which would make it possible for the driver to be working in normal clothes regardless of the zone in which he is operating. If properly analyzed in the beginning and the evolution toward this end taken in proper stride, there is no reason to suspect that we shall be called upon to pay an inordinate price for such creature comforts as seem so eminently desirable in the vehicles of the future. Many of these same things, if the price can be held within reason, will be used as the standard regardless of the service, as they will apply to a large degree equally to the city driver as well as the over-the-road driver.

Also, there is an urgent need for designing instruments better suited than those now used. These fatigue reducing features which have been described will be and should be largely automatic with simple controls to make the necessary adjustments in seating and ventilation.

Materials which we hope will be available in the future, in addition to being corrosion-proof, will have insulating properties as an in-built feature of the material itself. These features, together with noninflammability and shock absorbing qualities, are well within the limits of the plastic field. The shape necessary for wind resistance as well as the need for maximum vision can be more readily achieved perhaps by the use of plastics. Unquestionably, the development of plastics in the passenger-car field will have a great influence on the possibility of this development in

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WARTIME REFINERY OPERATIONS

and Their Effect upon

MOTOR and AVIATION FUELS

by W. M. HOLADAY
Socony-Vacuum Oil Co., Inc.

THE petroleum industry has done a pretty fair job of publicizing the fact that the armed forces of the United States, indeed the armed forces of the United Nations, were not lacking at any time during the course of the war for petroleum products to operate their war machines. Many specific statements have been made indicating spectacular achievements by the industry, foremost among which, perhaps, is the one indicating that high-octane-number aviation gasoline production was expanded over 15 times between December, 1941, and August, 1945. At the completion of the war we were producing 100-octane-number aviation fuel at a rate in excess of 500,000 bpd. Potential productive capacity for lower grades was in excess of 150,000 bpd. For purposes of comparison, in a normal prewar peacetime year, our needs were amply satisfied with only 5000 bpd.

Of more interest to the reader is what effect wartime operations, including the installation of new refining equipment, will have upon the quantity and quality of products that will be available in the postwar period for operating automotive and aircraft equipment. Comments will be made on aviation fuels, motor gasolines, tractor fuels, diesel fuels, and gas turbine fuels. Brief references will also be made to domestic heating oils and lubricating oils.

[This paper was presented at a meeting of the Milwaukee Section of the SAE, Milwaukee, Wis., Feb. 1, 1946.]

THE effect of wartime refinery operations, including the installation of new refining equipment, on the quality and quantity of products that will be available for operating automotive and aircraft equipment is fully discussed here by Mr. Holaday.

The author's comments cover aviation fuels, motor gasolines, tractor fuels, diesel fuels, and gas turbine fuels, as well as domestic heating oils and lubricating oils.

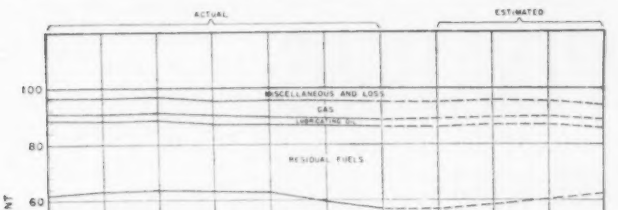
THE AUTHOR: W. M. HOLADAY (M '28), who has contributed many technical articles to the *SAE Journal*, has been in the petroleum field for almost 20 years. In 1944 he was appointed general manager of the Research & Development and Technical Service Laboratories of Socony-Vacuum Oil Co.—his present position. Immediately before, he had served for one and one-half years with the Petroleum Administration for War as chief of the Aviation Section, and later as assistant director of refining. This interval interrupted an association with Socony-Vacuum that started in 1937. A graduate from Ohio State University in 1925, Mr. Holaday was employed at Westinghouse Air Brake Co. for two years, after which he spent a 10-year period with Standard Oil Co. as automotive research engineer.

■ Yields of Principal Products—1937-1947

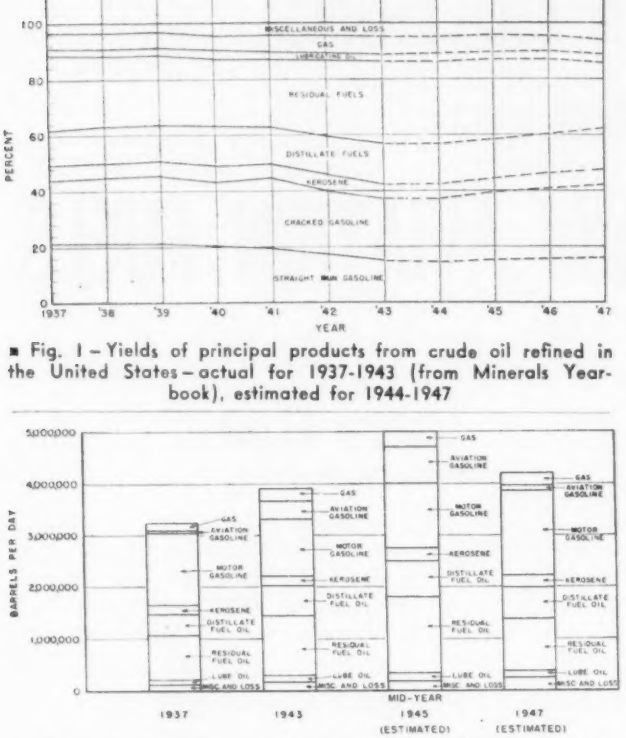
For background information refer to Fig. 1, indicating the yields of principal products from the crude oil processed in refineries in the United States from 1939 through 1943 upon which operations actual figures are available. We have gone on from this point, estimating yields during the last period of the war and during the years 1946 and 1947.

Yields of the principal products from petroleum in the prewar years covered remained substantially constant. These were about 43% of gasoline, 5% of kerosene, 14% of distillate fuels, 25% of residual fuels, 3% of lubricating oil, and 10% miscellaneous products, including waxes, road oils, asphalts, gas, and loss. There was a slight trend during this period toward increased cracked gasoline in the total motor fuel produced and a corresponding decrease in residual fuels.

With reference to yield, the principal effect of the war economy was to increase yields of residual fuels primarily for the operation of increasing amounts of naval and marine equipment and to satisfy increasing demands of industry. These increases in yields of residual fuels were made largely at the expense of gasolines. Present estimates on trends in the immediate future indicate a course back toward relative yields being experienced just before the war. If Fig. 1 were carried on toward 1950, the outlook should be for increased production of distillate-type fuels and gasoline, with proportional decreases in yields of residual fuel. Such a trend would appear to be indicated with increasing demands for the distillate-type product in home heating equipment and in internal-combustion engine equipment, both of the diesel and gas turbine types. These figures indicate gasoline yields slightly below prewar levels. We do not anticipate that the use of the internal-combustion gasoline engine will decline to any great extent but economy characteristics should be improved, so that with increased mileage per gallon, more cars will not necessarily burn more gasoline.



■ Fig. 1 – Yields of principal products from crude oil refined in the United States – actual for 1937-1943 (from Minerals Yearbook), estimated for 1944-1947



■ Fig. 2 – Crude runs and yields of refined petroleum products in the United States – 1937 and 1943 actual (from Minerals Yearbook), 1945 and 1947 estimated

Fig. 1 gives only a part of the story, inasmuch as it makes no reference to relative quantities of crude petroleum processed during the prewar period, the war period, and those anticipated in the immediate future. Fig. 2 gives this information.

In 1937 the petroleum industry in the United States was processing about 3,400,000 bpd of petroleum. As indicated in Fig. 2, motor gasoline production at this time amounted to about 1,400,000 bpd and a very small amount of aviation fuel.

By 1943 crude runs had increased to 3,900,000 bpd. As indicated in Fig. 1, residual fuels had increased out of proportion to the increase in crude run at the expense of gasoline. The total volume of motor gasoline had decreased rather substantially, largely at the expense of aviation gasoline production which had increased nearly ten-fold in the period 1937 to 1943. It was during this period that the supply of motor gasoline for public consumption was the tightest. Residual fuels showed their remarkable increase volume-wise through utilization of heavier crudes, containing little gasoline and light fuel oils, which could be satisfactorily processed on old refinery equipment, much of which had been shut down prior to the war.

The first and second quarters of 1945 exemplify maximum output by the petroleum industry. During this period 5,000,000 bpd of crude was being processed. Here we find marked increases volume-wise in gasolines, almost entirely attributable to the increase in aviation fuel output which approached 700,000 bpd against a motor gasoline production of 1,240,000 bpd. Probably 20% of this motor fuel went to the armed forces. It is noted that there is also a further increase in residual fuel oil production growing from slightly over 1,100,000 bpd in 1943 to 1,450,000 bpd at midyear, 1945.

Predictions would indicate that in 1947 refinery runs will amount to about 4,200,000 bpd of crude oil and, as indicated previously, the trend in yields of the various products will be back toward prewar levels. Motor gasoline production will be back to the range of 1,600,000 bpd, as compared with less than 1,200,000 during the war period, with a drop in aviation gasoline production to around 100,000 bpd. This increased yield will be at the expense of residual fuel oils. The trend will be toward higher cracked gasoline content in the total motor fuel as a result of the use of catalytic cracking equipment installed during the war period.

With crude runs and yields of products as background, let us review briefly the refinery processes and equipment that are involved and the changes that were required to meet the war demands.

■ Refinery Operations

Fig. 3 outlines in block diagram form the refinery processes used for manufacturing fuels. Starting at the producing field, the crude is stabilized, separating gaseous hydrocarbons which refuse to stay in solution in the crude under atmospheric pressure and temperature conditions. This gas is compressed and sent through an absorption system to remove a natural gasoline for subsequent blending with motor gasoline and to segregate butane and propane fractions for liquefied petroleum gases from light residual gases which are either burned directly as fuel in the refinery or oil fields or returned to the ground to repressure oil sands. The stabilized crude is then subjected to a primary distillation step yielding additional butane-propane fractions, part of which is used in subsequent alkylation processes; straight-run gasoline, naphthas and solvents; light fuel oils, which may consist of kerosene, diesel fuels, and so-called No. 2 fuel oil; lubricating stocks and heavy fuels. A portion or all of the light and heavy fuel stocks may then be subjected to either catalytic or thermal cracking or both. Catalytic cracking in general is limited to the utilization of lighter fuels than thermal cracking. Both of these processes yield, first, very light gases composed primarily of methane and ethane-ethylene, which are generally used for refinery fuel. (Ethylene, however, promises to be an important source of a multitude of chemicals derived from petroleum.) Second, there are other gaseous butane and propane fractions, a portion of which may be polymerized or alkylated and the remainder, generally butane, used to pressure the finally blended motor fuel. Third, there are substantial yields of cracked gasoline ranging from 25 to 50% of the total charge, the relative quantity depending upon charge stock to the process and the cracking conditions. Fourth, there are cracked, light fuel oils which are blended with straight-run products for domestic use; and last, heavy fuel oil, tar or coke.

■ New Refining Equipment

The major portion of the new equipment that was installed during the war period was constructed primarily to augment aviation gasoline production. It is this equipment that we must evaluate concerning the effects upon postwar production of commercial fuels and other petro-

leum products. Fig. 4 serves to indicate how aviation fuel production was expanded at a phenomenal rate, and the part the new facilities played in that expansion. It will be noted that until about the end of 1943, increments in production were obtained primarily through bottleneck removals, the conversion of the few catalytic cracking units that were in operation prior to the war from motor gasoline production to aviation-base stock production, and by the conversion of polymer units, which before the war manufactured motor gasoline, to operations for codimer and cumene production. Additional increments were obtained through relaxing specifications on the fuel itself. Beginning early in 1944, major components of new construction came into production.

The foundation stone in all of this production from new units was catalytic cracking. This operation provided superior quality base stocks from the point of view of antiknock quality for blending with aviation alkylate, and it also provided butylenes and some of the isobutane for the synthesis of the alkylate. Complementary to the catalytic cracking facilities were alkylation units themselves and isomerization units which converted normal butane into isobutane, which reacts with the butylenes in the alkylation unit to make aviation alkylate.

By 1945, the capacity of catalytic cracking facilities was 900,000 bpd of charge stock. The major portion of the material fed to these units to produce high-quality gasolines consists of distillate fuel oil stocks whose alternate uses are high-quality burner fuels of the domestic and light industrial types, and high-quality diesel fuels. During the last year of the war, the petroleum industry was hard-pressed to meet the competing demands for straight-run fuel oils for catalytic cracking and ultimate aviation fuel production on one hand and the direct requirement for that product as diesel fuel on the other. Only minor quantities of uncracked distillate found their way into domestic and light industrial fuels, making those products somewhat less adaptable to the equipment using them. Difficulties arose in maintaining clean combustion and in avoiding carbon deposition in burners and boilers. Thus newly installed refinery equipment has for the last five years been constructed with the view of making light products of high antiknock quality at the expense of petroleum fractions in the fuel oil range, and the conversion of gaseous petroleum fractions into high-octane-number liquid fuels.

Aviation Fuels

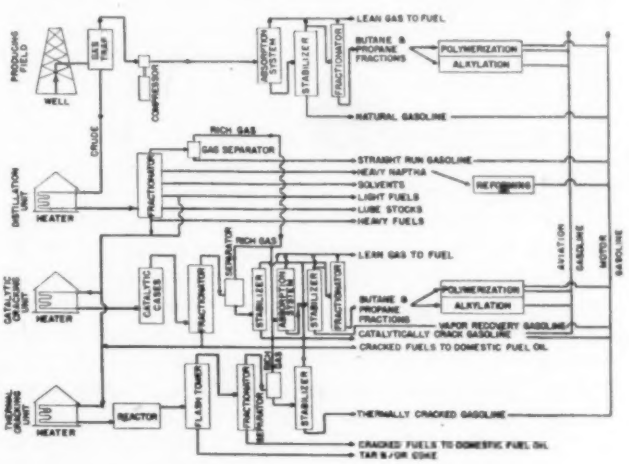
As an introduction to the quality of aviation fuels that can be expected within the next few years, let's review briefly what was produced during the war. During this period, the armed services utilized six grades of aviation gasoline. Each was designated by its antiknock quality or octane number. These were grades 62, 73, 80, 87, 91/96, and 100/130. Just at the close of the war a seventh grade, the so-called super-quality grade 115/145, was being produced at several refineries in this country.

Grades 91 and lower were generally straight-run products containing increasing amounts of lead as the octane-number grade increased. Grade 62 was used primarily for engine run-out, for cleaning leaded fuels out of an engine following initial tests and prior to some indefinite storage period for the engine. Grades 73 and 80 were used in primary training. Grade 87 was also used in some specialized training activities and as a substitute for grade 91/96 in some of the less critical operations generally requiring the higher grade. Grade 91 was used by commercial aircraft and by Army and Navy transport equipment operating in this country.

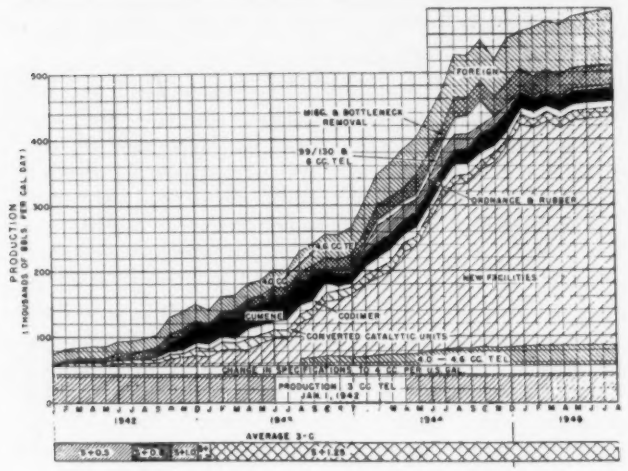
Grade 100/130 was generally composed of alkylate, isopentane, catalytically cracked base stock of high antiknock quality, some straight-run base stocks, and relatively small percentages of aromatic components to attain the required rich mixture quality. This fuel, containing 4.6 cc tel per gal, was used exclusively for combat purposes and for transoceanic and long-distance transport operations. Fig. 5 shows the trend in average composition of 100-octane-number fuel, by year, from 1941 to 1945.

Grade 115/145, which was manufactured at the end of the war but never utilized in combat theaters, was destined for combat use. This fuel contained a very high percentage of alkylate, some isopentane, and generally only a relatively small amount of selected catalytically cracked base stock. It likewise contained 4.6 cc tel per gal. A typical blend of grade 115/145 is compared with grade 100/130 composition-wise in Fig. 6. With the availability of aviation alkylate limiting all-out production of aviation fuels in the octane-number range of 100 and higher, it may be seen that the potential output of grade 115/145 was only about half that of grade 100/130.

During the war, with all-out and in many cases uneconomic refinery operations, aviation alkylate production reached a maximum figure of about 170,000 bpd. This material, when containing 4.6 cc tel per gal, is superior to the grade 115/145, the so-called super-quality aviation gasoline. Now if we look at anticipated postwar requirements for aviation fuels and note that they are established at about 100,000 bpd, it is apparent that adequate quantities of superior quality fuel can be made available. It should be understood, however, that eliminating uneconomic operations, and on the basis of reduced crude runs, potential output of alkylate will probably be reduced to the range of 100,000 bpd. Further, it should be understood that super-quality fuels will cost more than the 100-octane-number grade. Two factors influence this: First, alkylation involves an additional processing step which cannot be considered inexpensive; and second, alternate application of the components used to make the



■ Fig. 3 – Refinery processes for manufacturing fuels



■ Fig. 4 – Actual production rate of 100-octane gasoline (worldwide) – 1942-1945 (by production increments)

alkylate must be considered. The alternate use for the butanes consumed in this process is pressuring motor fuels. In this case their value is augmented by their relatively high antiknock quality, around 95 octane number. In addition, it takes roughly two volumes of butanes to make 1.2 volumes of alkylate.

Immediately after the war, interest was again aroused in the use of distillate types of aviation fuel with a flash point above 100 F, and distilling in the range of 300 to 400 F. The original designation for this product was aviation safety fuel. However, after arguing for 10 years on "how safe is a safety fuel," a move is now on to designate it as low-volatile aviation fuel or low-volatile fuel.

Some airline operators, and to a considerably lesser degree, engine builders, have considered the use of such a fuel for a number of years with the view toward reducing fire hazards in fueling and in minor crashes. Its use was indicated to be impractical, however, until fuel-injection equipment was developed and adapted to aircraft-engine equipment. This was accomplished in the larger horsepower units used late in the war period.

Refining equipment installed during the past five years has placed the petroleum industry in a position to supply reasonable quantities of low volatile fuel in the 100-octane-number range, should the demand arise. Materials available against such a potential requirement fall into two categories with reference to the type of processing from which they are derived and their general chemical characteristics. Alkylation units can be made to operate to produce suitable materials in conjunction with regular, light alkylate. This product would be isoparaffinic in nature, as would the material resulting from the hydrogenation of trimers from the selective polymerization of butylenes. This type of product would be characterized by a relatively high heating value, generally in the range of 18,800 Btu per lb, and usually would be described as nonsensitive relative to engine operating conditions. A second general source is catalytic cracking. Special processing steps would be required, such as catalytic retreatment or hydrogenation. This product would be aromatic in nature and thus would have a lower heating value per pound. It would also tend to be more sensitive to engine operating conditions.

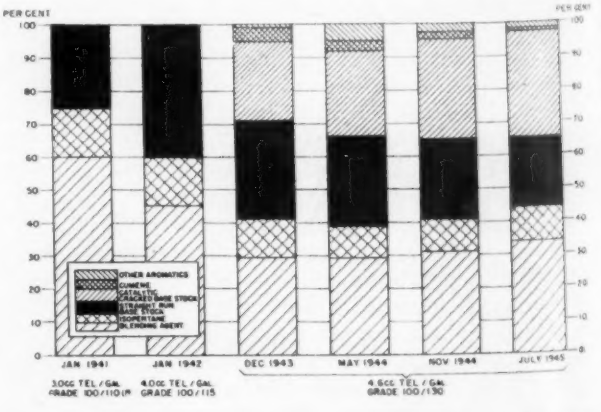
Unfortunately, in the past few months potential consumers have tended to raise their sights on antiknock quality and are now seeking low-volatile fuels in the range of 115/145. In the present state of development of refinery processing, such quality would seem to be out of the economic range of fuel costs to airline operators.

■ Motor Gasolines

With that brief review of aviation fuels, let us now look at the quality of motor gasolines that we can expect to have in the course of the next five or six years. The dwindling demand for aviation gasoline has made available to motor fuels substantial quantities of high antiknock quality gasolines resulting from catalytic cracking. These catalytically cracked products are not as high in quality as they were during the war period for aviation gasoline blending. In order to obtain both stability and the highest possible octane number, this material generally had to be either retreated catalytically, subjected to hydrogenation, or a rather drastic acid treatment. The first procedure eliminated olefins and increased aromatic content; the second merely saturated olefinic compounds which display varying degrees of instability, and the third removed olefinic constituents. Such additional treatment naturally added to costs, and where catalytic retreatment was required, had the effect of reducing throughout on the catalytic cracking unit. Consequently, although antiknock quality characteristics will not be as high as during the war period, this factor should be partially offset by increased volumes.

The effect of the diversion of these stocks from aviation gasoline to motor gasoline is already apparent in the marked increase of antiknock quality of motor gasolines marketed at the present time over that marketed in the prewar period. In some areas of the country, this increase amounts to four to five octane numbers. On the average, perhaps it is between three and four. A careful analysis of motor fuels inspection data will reveal, however, that at least a part of this increase in antiknock quality, probably between one and two octane numbers, has been obtained through the use of more tetraethyl lead per gallon of finished motor fuel.

It seems to be the consensus that the upper limit in the present range of octane numbers of motor fuels has become more or less stabilized and probably will not change



■ Fig. 5 – Composition of 100-octane aviation fuels – 1941-1945

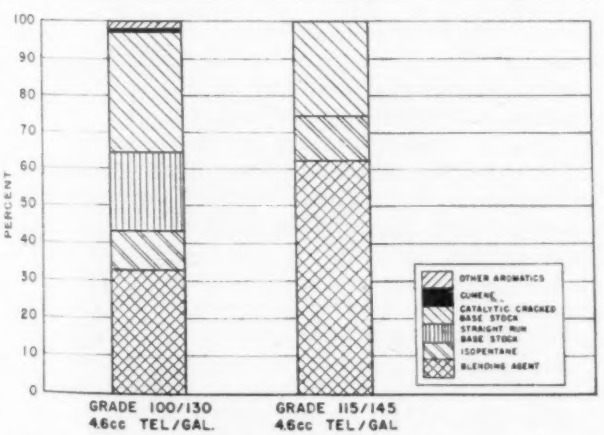
appreciably for some indeterminate future period. Premium grades, with Motor Method octane numbers in the range of 79 to 85 and Research Method octane numbers in the range of 84 to 93, are being marketed throughout the country. Quality on the East and West Coasts in this respect is generally higher than throughout the Midwest. This same statement applies to regular-grade gasolines which have Motor Method ratings in the range of 73 to 80, and Research Method octane numbers in the range of 77 to 86.

Concerning other characteristics of the motor fuels, it is anticipated that vapor pressures may increase somewhat and that distillation ranges will likewise increase, keeping overall volatility characteristics equivalent to the prewar level. For a number of years, fuel systems in automotive equipment have been gradually improved to utilize in a satisfactory manner higher vapor pressure gasoline, and as a result, more light fractions will be utilized in motor fuel from economic and conservation standpoints. Catalytic cracking has provided not only additional butanes for vapor pressure control but also fractions boiling high in the gasoline range having superior antiknock quality. Road tests in automotive equipment indicate that these heavier fractions can be advantageously utilized from the economy-mileage standpoint without danger of excessive lubricating oil dilution or adverse effects upon starting and warmup performance.

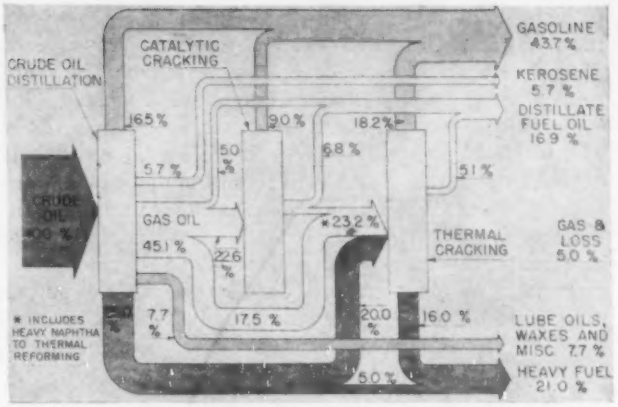
To summarize, it appears that the engine builders and the gasoline consumers can count upon the availability of fuels in the range of four octane numbers higher than those that they used before the war and fuels with similar overall volatility characteristics, although there will be increased utilization of both extremely light fractions and heavy fractions in the gasoline. Further modest increases in antiknock quality are possible through the use of still more tetraethyl lead. Beyond that, some rather radical changes in refinery operation will be necessary to increase octane number.

Diesel Fuels and Gas Turbine Fuels

Predicting the effect of wartime refinery construction on the quality of diesel fuels and gas turbine fuels that will be available is basically a question of economics. Unfortunately the quality characteristics of a good diesel fuel, a good domestic burner fuel, and, apparently, at the present stage



■ Fig. 6 – Composition of 100/130 and 115/145 grades of gasoline



■ Fig. 7 – Estimated postwar refinery operation for all the United States

of engine development, a good gas turbine fuel are identical to those of a good cracking stock. These are found in the distillate fuel taken directly from the crude. In view of the fact that wartime refinery construction involved mostly catalytic cracking requiring such distillate fuels for optimum operations, the question is put to the refiner: "Where should I put my straight-run distillate to attain the most economical operation?"

Fig. 7 shows a schematic flow diagram of a refinery using the three major refining operations, namely crude oil distillation, catalytic cracking, and thermal cracking and illustrates wherein the refiner's problem lies. The fraction denoted as gas oil from the crude distillation step represents about 45% of the total crude and contains all of the light distillates suitable for internal-combustion engines of the diesel and gas turbine types. You will note that in this estimated postwar refining operation, 90% of this fraction is charged to catalytic and thermal processing, with a resultant yield of gasoline amounting to over 60% of the refinery's total output of that product. 28% of the initial distillate fuel from the crude oil has been converted to other products. Only 5%, based upon crude, or 30% of the total distillate fuel oil is utilized directly in the final product.

In order to ensure the highest quality diesel fuels, the refiner must segregate his requirements for this product from the remainder of the distillate fuel, which goes largely to commercial and industrial consumers as burner fuel. To make the problem a little more difficult, diesel fuels of satisfactory ignition quality are not available from all crudes that a refiner may process. Consequently, further segregation may be necessary.

Assuming decreased gasoline demands or a premium in price on straight-run distillate fuel oils, economics allows the refiner to segregate more of this material and still keep his catalytic cracking unit and thermal cracking unit running to capacity by recycling distillate materials resulting from these processes. Under present conditions the economics of this procedure of operation is not attractive inasmuch as distillate fuels from cracking operations are refractive, giving poor yields of required products under more stringent cracking conditions which are costly from the maintenance standpoint, and result in increased conversion of liquid products to coke and gas having considerably less value.

The point that we wish to make is that diesel fuels and gas turbine fuels of good quality can be made available

in increasing quantities, but the effect would be to reverse the trend of refinery technology during the past 10 years. The installation of catalytic cracking equipment has improved gasolines for the spark-ignition engine. This improvement has been requested by the manufacturers of this type of equipment. With it has come a concurrent degradation of the distillate fuels remaining for compression ignition and gas turbine engines of the current design. Refinery operations involving catalytic and thermal cracking equipment may be altered to reduce residual fuel yields, utilize more cracked distillate fuels, and release straight-run distillate but at substantially increased operating costs. It would appear that these costs must be paid by the user of selected distillate fuels.

Reference has been made to gas turbine equipment. This is a new field, and we should not have too many preconceived notions concerning fuel quality requirements. The only significant demands for a product to fulfill this type of requirement that have arisen are in connection with military jet-propelled aircraft. It is possible, indeed probable, that this application would be most delicate, as is the case of internal-combustion spark-ignition engines for aircraft use. Nevertheless, it has been established that presently designed jet engines have to be fed a rather carefully specified diet to avoid carbon deposition and to ensure flame stability. Straight-run fuels, paraffinic in nature (high hydrogen to carbon ratio) are required. For best economy and range, these must be in the fuel oil classification. Development of stationary, locomotive, and marine gas turbines should be undertaken with the view toward using cracked distillates or better, residual fuels, for optimum availability and best economy from the fuel cost standpoint.

■ Tractor Fuels

A brief statement might be made concerning general availability and quality of tractor fuels of the distillate type. In this application, catalytic cracking is probably being of assistance. As mentioned previously, this type of processing yields fractions high in the boiling range of gasoline, in fact that constituting the lighter portion of the distillate fuel, which have good antiknock properties. Such catalytically cracked material, either alone or as a blending agent, should serve generally to increase the level of antiknock quality of distillate tractor fuels.

Cooperative activities in establishing tractor fuel classifications are healthy signs and, if augmented by a program for the education of the tax authorities, tractor equipment operators should experience a happier lot with more uniform and better distillate fuels with reference to both antiknock quality and overall economy of operation.

■ Domestic Heating Oils

Although not falling strictly within the scope of the subject of this paper, a few general remarks about fuels used in domestic oil burning equipment may be of interest. We have already alluded to the fact that high hydrogen to carbon ratio (straight run distillates) ensures cleanest burning in certain critical types of domestic equipment. Catalytically cracked fuel oils are typified by substantially lower hydrogen to carbon ratio, and if burning difficulties are experienced due to carbonization of nozzles and ignitors and deposition on heat transfer surfaces, trouble will result from the use of this type of fuel sooner than it will result from the use of straight-run fuel. However, if such catalytically cracked material can be used, it affords, with its lower hydrogen to carbon ratio and resulting lower specific gravity, greater heat content per gallon and thereby greater economy for the consumer.

In general, atomizing types of oil burners can successfully use catalytically cracked distillates. On the other hand, certain vaporizing types are found to suffer from carbon deposition and resultant inefficient operation. This inherent weakness in some types of vaporizing equipment has been recognized by manufacturers. Some improvements have been made in combustion characteristics, and it is anticipated that more will be made in the future. Until such time as these problems can be completely overcome, a most select fuel, such as kerosene, must be used in much of the equipment that falls into this classification.

The availability of kerosene always has been, and perhaps will continue to be, quite limited. Since domestic heating equipment of the vaporizing type became popular in the middle 30's, refiners have tended to produce the maximum amount of kerosene available in the crudes processed by them. In passing, it should be understood that kerosene can be obtained by simple distillation only from paraffinic crudes, and mixed-base crudes which contain predominantly paraffinic constituents. "Kerosene" fractions from other crude sources require special treating steps to meet burning quality requirements that have been established for this product. It demands a premium in price over other distillate fuels for the reasons that its production is limited, it requires special handling and treating in the refinery, and distribution costs are in general higher because of the substantially smaller volumes moved in comparison with gasoline and the more widely used fuel oils.

■ Distribution Costs of Petroleum Products

Equipment manufacturers should give due consideration to distribution costs of petroleum products in the course of the design of new units. To keep down ultimate cost to the consumer, it is essential that the manufacturer do his utmost to make his equipment operate in a satisfactory manner on those petroleum products which are in greatest demand and handled in largest quantities. Commodities such as gasoline and domestic heating oils can be readily handled in tankers in coastal areas and by pipeline in inland areas because of the large volume of individual shipments. On the other hand, fuels with special physical or chemical requirements must be moved in tank cars or in drums. Distribution costs may thereby be increased as much as several hundred per cent.

These distribution costs on products handled in small volume are particularly significant in the Middle West with reference to equipment operated on farms. With the population and general demand scattered as it is, main storage depots are far from the ultimate consumer and delivery involves long and devious routes. The same general comments likewise apply to aviation fuels with airports of the emergency and resort types situated often in rather inaccessible spots. The consumer will benefit by every move that the manufacturer can make to have his product operate satisfactorily on those petroleum fuels which are in greatest demand.

The effect of wartime development on the quality of lubricants likewise is somewhat out of the scope of this

paper but deserves passing comment inasmuch as advances in engine operating efficiency through improved fuels will probably impose added lubrication problems. The extensive use of motorized equipment during the war, and the urgent requirement that it operate perfectly under the most grueling conditions, accelerated research activities resulting in increased knowledge of basic lubrication requirements and markedly improved products. These advances were due principally to the development of oil additives to improve base oils in specific directions; that is, detergency, oxidation stability, oiliness, e-p characteristics, and so on.

Conclusion

Cooperative activities on the part of equipment builders, petroleum refiners, additive manufacturers, and the armed services lent in no small way to the satisfactory solution of the problems arising during the war period on fuels and lubricants. Through this cooperative action, problems became more clearly defined by wider dissemination of basic information, responsibility for difficulties could be more readily established, and limitations on the part of the groups concerned to remedy difficulties were better understood. All of these advances are now applicable in a large measure to improvements in the service of fuels and lubricants in peacetime applications.

If there is anything that can be said in favor of war, it is the fact that during such a period enormous advances are made in science and technology which can be readily and rapidly applied during the following years of peace. Research and development are always geared at a higher rate, and the time of application between development and general utilization is decreased many-fold. In this speedup some wrong and costly paths are traveled that are justified only under a war economy. Cooperative activities, on a scale found only in wartime, have had the effect of throwing road blocks across these paths that come to a dead end. Competition and individual initiative will eliminate a good deal of this cooperative effort under our present economy, but sufficient activity of this type must be continued among industry for rational application of wartime advances to a peacetime economy and for continued rapid development of better products to most efficient uses.

DEVELOPMENT and APPLICATION of HEATED WINGS

continued from page 269

2. NACA ARR, July, 1941, "Flight Investigation of Thermal Properties of Exhaust-Heated Wing-De-icing System on Lockheed 12A Airplane," by L. A. Rodert and others.

3. NACA ARR Supplement No. 1 to above report, July, 1941.

4. NACA ARR Supplement No. 2 to above reports, September, 1941.

5. NACA ARR, January, 1942, "Recent Flight Research on Ice Prevention," by L. A. Rodert and others.

The above reports deal with flight experiences of the Lockheed 12A airplane when flown in icing conditions. Experimental data in dry air and in ice are presented.

6. NACA ARR, April, 1942, "Preliminary Investigation and Design of Air-heated Wing for Lockheed 12A Airplane," by L. A. Rodert and others. A preliminary analysis of the use of heat exchangers and hot air in an anti-icing system.

7. NACA ARR, October, 1942, to 1945, "Investigation of Aircraft Heaters," Sections I to XI by L. M. K. Boelter and others. These reports contain general information on heat transfer as applied to aircraft. Tests of several heat exchangers are correlated to empirical formulas. Properties of gases, radiation, a general treatment of aircraft heating and ventilating problems.

8. NACA ACR, February, 1943, "Development of Thermal Ice Prevention Equipment for B-24D Airplane," by L. A. Rodert and others. A comprehenseive analysis of the design problems in heated wings. The B-24 is used to demonstrate the method of calculation employed in designing the NACA ice research airplanes.

9. NACA ACR "Tests of Heated Low-Drag Airfoil."

10. NACA ACR, December, 1942, "Method for Determining Rate of Heat Transfer from Wing of Streamline Body," by C. W. Frick and G. B. McCullough.

These two reports contain an excellent theoretical treatment of the heat-transfer and drag properties of heated airfoils. Well substantiated by wind tunnel data.

11. NACA Restricted Bulletin, December, 1942, "Method for Calculating Heat Transfer in Laminar Flow Region of Bodies," by H. J. Allen and B. C. Look. Theoretical treatment of the laminar flow region about airfoils or other streamlined bodies. Essentially the same treatment as that presented by Frick and McCullough in above reports.

12. NACA Technical Memorandum No. 1045, May, 1943, "Heat Transfer to Plate in Flow at High Speed," by E. Eckert and O. Drewitz.

13. NACA Technical Memorandum No. 1044, April, 1943, "Heat Transfer of Airfoils and Plates," by O. Seibert. Experimental data with a correlation to empirical formulas.

14. NACA Technical Memorandum No. 1027, August, 1942, "Formation of Ice on Aircraft," by W. Bleeker.

15. NACA Technical Memorandum No. 1028, September, 1942, "Icing of Aircraft," by M. Robitzsch.

16. NACA Technical Memorandum No. 1000, December, 1941, "Temperature of Unheated Bodies in High-Speed Gas Stream," by E. Eckert and W. Weise.

17. NACA ACR, May 8, 1943, "Flight Tests of Thermal Ice Prevention Equipment on Lockheed 12A Airplane," by L. A. Rodert and others. Test data on the air-heated system in both wet and dry air.

18. NACA reprint of RAE Technical Note No. Inst. 674, June, 1942, "Note on Kinetic Heating with Particular Reference to Conditions of Icing." A discussion of aerodynamic heating in icing conditions. Comparisons to experimental flight data.

19. NACA reprint of RAE Report No. Aero 1783, November, 1942, "Heat Transfer Calculations for Aerofoils," by H. B. Squire. A general discussion of heat transfer for airfoils and a method of performing the necessary calculations. Comparisons are made to some experimental data.

Discussion of

THE paper "Engineering of Involute Splines," by George L. McCain, was published in the May issue of the SAE Journal (Transactions) on pp. 245-260.

Included herewith is the very extensive written discussion that was read following the presentation of Mr. McCain's paper.

This discussion, in effect, supplements the main paper, contributing additional information on production problems, effect of interference and index errors and misalignment, broaching as applied to the new standard, and the use of a single radius in place of the involute curve.

Advantages Shown For New Standard

— CHARLES H. STANARD
Buick Motor Division

IT was in January of 1905 that I milled 10 sliding gear transmission main shafts and a short square broach to finish the square hole in the malleable sleeve in which the low and reverse gear, and the second-speed and direct clutch gear were riveted. Nice round clutch teeth were not to come for several years. These were 3-jawed dog clutches, backed off, and with ⅛-in. clearance or backlash. The multiple disc clutch hadn't arrived to plague our sleep yet. They were just cone clutches and we didn't make them. (See Figs. 1 and 2.)

During the period from 1905 to 1913, the transmission was gradually changing. The dog clutches had been superseded by the round clutch teeth and the cone clutch had been generally replaced by the multiple disc clutch with riveted keys. The spline shaft had started to supersede the square shaft. (See Fig. 3.)

From 1913 to 1920 the pressure was for production and design was changing toward larger and heavier units. Advance in machine and tool design had been faster than in transmission design. The ground hob and cutter and the grinding of gear teeth had made possible a tremendous gain in accuracy of product.

The company where I worked had tried out the straight splined shaft and found many reasons why the old square shaft should remain as the main shaft. If customers demanded that the transmission be equipped with spline shafts, it was. All the care possible was used in their manufacture but with little satisfaction to the customer or maker.

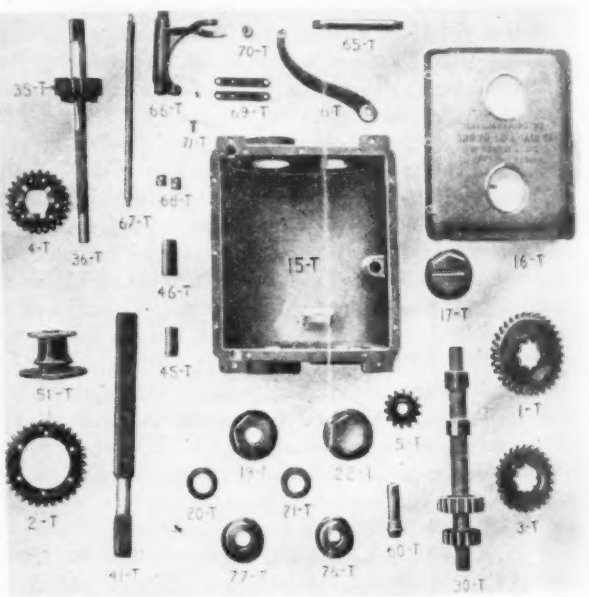
Many suggestions were offered during this period for refinement of fittings, but if they were out of the ordinary they were not salable.

I like to look back upon the period from 1920 to 1930 as the informative period. Tools were being designed and manufactured so that accurate measurements to tenths of thousandths could be made. The influence of hardening could be accurately explored and the inaccuracy of supposedly good tools could be seen and corrected.

The ability of the shopmen to inform themselves and the designer, coupled with Hamilton's paper[1] in 1920, whose facts were readily proved in manufacture, led to the adoption of the involute form for the gear clutch teeth as well as the fastening and positioning of the disc in the multiple disc clutches.

In 1927 I made the acquaintance of a designing engineer who thought much as I did about the import of shop problems, and in June, 1929, I went to work for and with him. I think he was the best designing engineer for production that I ever worked with. We designed and built a helical gear synchronizing transmission with involute splines. Circumstances made us redesign the job and we ended with an involute spline under the low-speed gear on the sleeve and in the clutches for high and second. After a year of production we placed the involute on the inside of the sleeve and on the main shaft. Later we were able to supplant the straight splined end of the main shaft with an involute form, but haven't been able to do the same with the clutch plate end of the clutch gear yet. (See Figs. 4 and 5.)

In 1939 the ASA B-5 1939 involute spline standard was



■ Fig. 1 – Progressive type of sliding gear transmission

[1] See *SAE Journal*, Vol. 7, August, 1920, pp. 149-154: "Methods of Generating Involute Spline and Its Application," by D. T. Hamilton.

"Engineering of Involute Splines"

approved. This system required the use of 76 basic hobs for the same reason that we still have straight-sided splines on the clutch gear and in spite of the facts that Mr. Hamilton had so ably pointed out in 1920.

The aircraft industry recognized the value of the involute formed spline and went ahead with a new simplified standard for its own purposes. Fortunately, heavy ordnance was able to use the involute to some extent, but seldom the standard.

This latest offering of an involute spline standard has been based upon ease of manufacture, control of quality, and functional value. It is a spline system of first-class engineering design and functional qualifications, as Mr. McCain has explained, but has been engineered for production.

■ Ease of Manufacture

Under ease of manufacture can be stressed the following points: The internal spline can be produced by a shaper cutter or a broach to a given green size. After hardening of the part, the fire change can be compensated for on the mating external spline. The external spline can be produced with a shaper cutter, formed milling cutter, or hob. If the part be hardened or heat-treated and growth or shrinkage affects the size and/or tooth form, this can be readily compensated for by changing the angle of sharpening of the tool and feeding deeper or shallower, as the case requires. Tools for locating a part having internal splines for successive operations can be produced with the tools used for producing the external splines.

■ Control of Quality

To control the quality of a part properly it becomes necessary to find out the various factors that may affect quality. This is readily done by using the various checking methods which have proved so satisfactory in the control of quality for production of spur and helical gears. It is important to know the accuracy of the profile, spacing, lead, concentricity, and thickness of tooth. It is safe to say that in too many cases, the failure to fit leads to a size change more often than the corrections of the factors causing the variation in size. It is very important to know the class of accuracy that one can expect from existing equipment and designs, so that the size tolerances can be adjusted to make satisfactory fits and still maintain the functional value.

Having followed closely the manufacture of splined parts, I am impressed with the need for recognizing and controlling the following factors in order that fitting qualification can be met.

■ Tooth Form

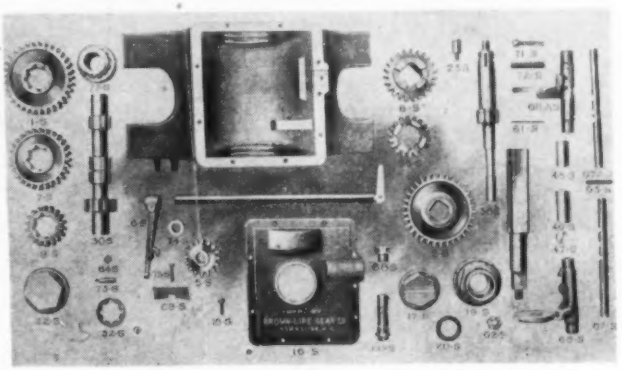
The tooth form of the internal teeth must be held to the basic pressure angle whether broached or shaped. Here the checking of sample products from the tools must be carefully made and tools held for tooth form. When the tools are made, it is important that the teeth shall be radial and not lean. If the tools are kept correct for tooth form, the normal movement of growth or shrinkage in heat-treatment can be determined and the teeth of the mating external spline be corrected for tooth form to fit. Hobs or cutters must be accurately centered for cutting external teeth in order that tooth form be symmetrical and not affect size.

■ Spacing

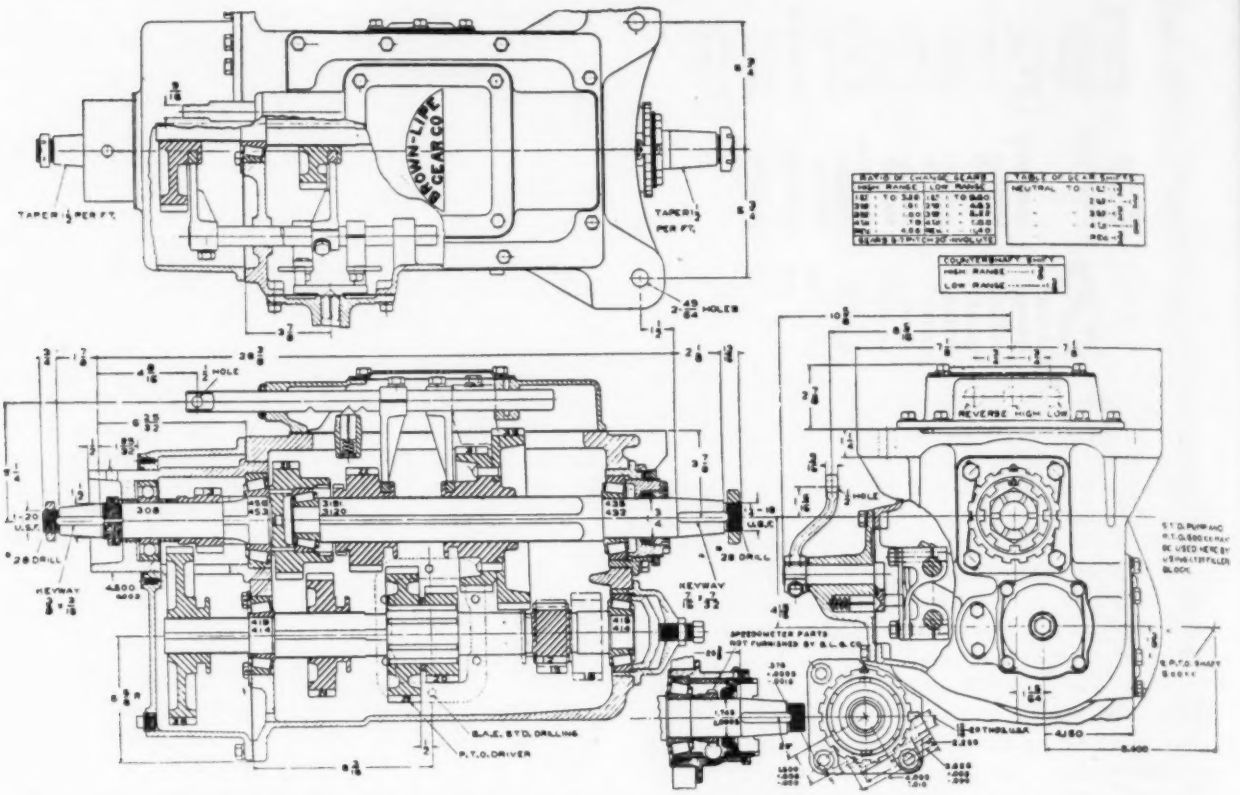
The teeth of the internal spline can be spaced very accurately if broached or shaped. Broaches and shaper cutters are made very accurately and the condition of the machine used in shaping will be the largest factor in spacing. It is necessary that the gear trains of the machine be kept in perfect condition to produce uniform angular velocity between the work and cutter. The external teeth will be most accurate if hobbed or shaped. Hobbing with single thread, ground form hobs will be better than when using multiple thread hobs. The hob manufacturer must have some manufacturing tolerance. If the number of teeth in the spline is not divisible by the number of threads in a multiple thread hob, the spacing will be best; although the tooth will be rougher. This roughness does not influence the size or functional value as much as spacing errors do.

■ Concentricity

The internal teeth are generally put in before the blanking is done and the part located from the spline. This is probably the most correct method, provided the holding tools do not engage upon the sides of the spline teeth but upon the major diameter. The major diameter as produced



■ Fig. 2 – Selective type of sliding gear transmission



■ Fig. 3 – Seven-speed main frame transmission with two-speed countershaft giving high and low range of four forward and one reverse speeds – overdrive on high range – overhead shift – power aperture – gears sliding on square shafts

by the broach or shaper cutter is more nearly circular and tools accurately contacting this surface will not deform or cold work the spline teeth. The external spline should be made by locating the part upon centers from which all operations are easily carried out.

■ Lead

The variation of parallelness of the tooth sides with the axis of spline parts must be held very exact. If a straight spline, both members must be straight or, if helical, both must be helical the same amount. Tremendous and thrust loads are set up between mating members if the splines are not conjugate with the axis. This results in slip out, excess bearing loads, noise, and other kindred troubles. Careful checking and immediate reconditioning of machines are important to remedy this trouble. For correcting normal fire change the lead of the external teeth is shortened or lengthened an amount to correct for the internal teeth, which are not generally corrected.

■ Mounting

The involute spline has been mistakably labeled as a side bearing spline and seemingly its entire use is predicated on this feature. It was not intended as such and its first main use was top bearing and so remains to date. A little study will convince the maker and user that it functions best as a side bearing spline when it is necessary to move laterally under load. For location and tight coupling it is best as a top bearing spline. The reason for this is easily seen. When necessary to move endwise under load, the spline acts as a gear, transferring this load at the point of force from tooth to tooth uniformly, maintaining a uniform position between the shaft and mating member. For locating or tight coupling where runout must be held to a minimum it must be remembered that both errors of tooth form and spacing in each member cannot be balanced; thus, the splines will locate upon three teeth with the position varying as a 3-legged stool on a cobblestone floor. Here again, the broach or shaper cutter will produce a major diameter in the fitting which is very nearly circular and the grinding of the major diameter of the shaft upon the centers on which the splines were generated will produce two surfaces whose areas will be greater than the contacts on the tooth sides. This makes the tooth sides conform and provides a fixed location regardless of the position of assembly of the spline teeth.

■ Fiber Structure

This last factor is, in my mind, the most important variable found in manufacturing splines or gears. We first isolated it in gear work and, as the involute spline is in reality a gear form, we have followed the influence of fiber structure through several troublesome designs. Perhaps we should make clear our mental picture of fiber structure by defining it as the arrangement of the lines seen in the etched surface of any section of a forging or rolled and formed piece.

Our success with the square shaft and square holed mating member was due to our broaching the square hole

to size, using it for location for all operations, hardening, and then just grinding out the fire change. The stock removed was at the ends of the hole under the hub and the fork groove and the wheel just shined the surface under the gear. We were not able to grind the internal straight sided spline and the fiber structure influence produced taper holes that could not be controlled.

Now it becomes necessary for us to come back to the designer and ask that he eliminate, in so far as possible, splines under light section, splines under hubs, and the design of forging so that we can finish with an accuracy that can be gaged and used. (See Fig. 6.)

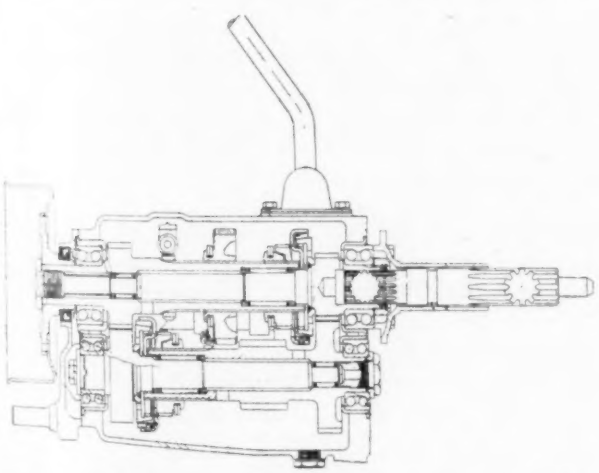
Figs. 7 and 8 show the fiber structure of a 1½-in. shaft that was offset ⅛ in. and end twisted through 45 to 90 deg. This shaft was heated and worked hot to duplicate forging practice, also normalized before having all evidence of offsetting or twisting removed by turning. Then four keyways were carefully cut on the quarter and checked for lead and width. Then the shaft was heat-treated, straightened, and ground on the outside diameter. Then the keyways were again checked and showed a pattern of change as shown in Fig. 9.

This movement may be caused by releasing strains when cutting across the fibers. The movement may not show itself immediately but will become apparent if part is shocked or even if left for a long period of time. This movement becomes apparent immediately upon heat-treatment and is charged as being caused by heat-treatment.

Fig. 10 shows an unwrapped spline on the inside of an aircraft-engine crank. The spline teeth were cut after all other work on the crank had been performed. The mating hub was to be installed with 1500 to 3500-lb. press fit. When made to the gages the parts went together with about 6-tons pressure. These splines were cut on the shaper and held to a pin size of 0.001-in. spacing to 0.0003-in. with tooth form, alike within 0.0002 in. There was 0.0008-in. tooth clearance instead of interference. The hub was assembled and removed in all units and a careful study showed a consistent pattern that followed the fiber flow of the forging.

Fig. 11 shows a pack hardened, press fitted, side bearing pinion spline. This shows that the tooth surfaces moved in relation to each other when hardened. It also shows that the members moved in relation to each other even under a press fit.

We have been asked to fix tolerances by which gages can be designed and their tolerances specified. This cannot be done with justice to the maker, user, or tool supplier. The problem of part design enters into the tolerance problem. My advice is to take the problem of design to the shop, the toolmaker, and metallurgist before setting up rigid requirements. If not, you can expect to fit the gages to the work or argue endlessly about them.



▪ Fig. 4 – Helical gear synchronizing transmission with involute splines

Effect of Interference Errors On Tooth Thickness of Splines

— ALBERT S. BEAM
Vinco Corp.

Practical experience with spline fittings frequently shows a discrepancy between actual tooth thickness and space width dimensions, as measured by means of pins or tooth calipers, and the backlash when the parts are assembled. It appears that the fit of two mating members is usually tighter than indicated by the difference between the space width of the splined hole and the tooth thickness of the splined shaft.

The reverse condition occurs when parts are inspected by means of spline gages with a full complement of teeth, the tooth thickness of which is equal to the specified space width dimensions of the parts. In these cases, the part splines appear to be looser in assembly than indicated by the gage dimensions.

These inconsistencies cause serious difficulties in the production and inspection of part splines. They are the result of spacing errors and other unavoidable deviations from the ideal spline contour.

In order to coordinate these facts with blueprint specifications recognition is required of the nature of these errors which interfere with nominal fit conditions. In this work, methods of identification and control are proposed, supported by a new terminology and a systematic analysis of inspection procedures.

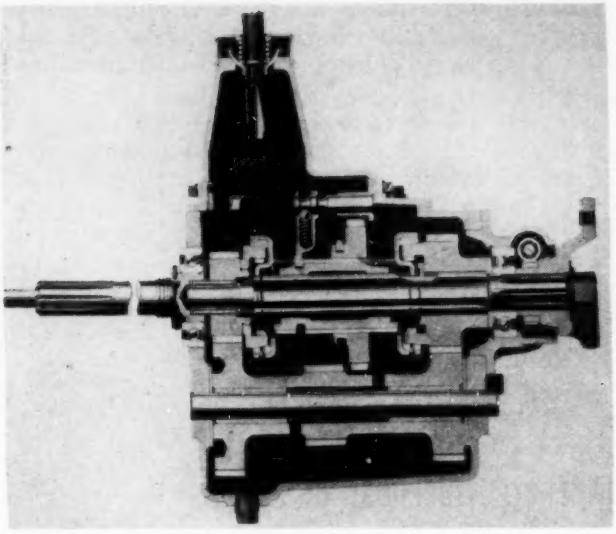
There are basically four errors which prevent splined shafts of a given tooth thickness from fitting splined holes of equal space width, as measured by means of pins. These errors are: spacing errors, profile errors, alignment errors, and roundness errors.

■ Interference Error

Inasmuch as these errors cause interference with the ideal spline contour, having the given tooth thickness at the pitch line, their combined effect is designated as "interference error."

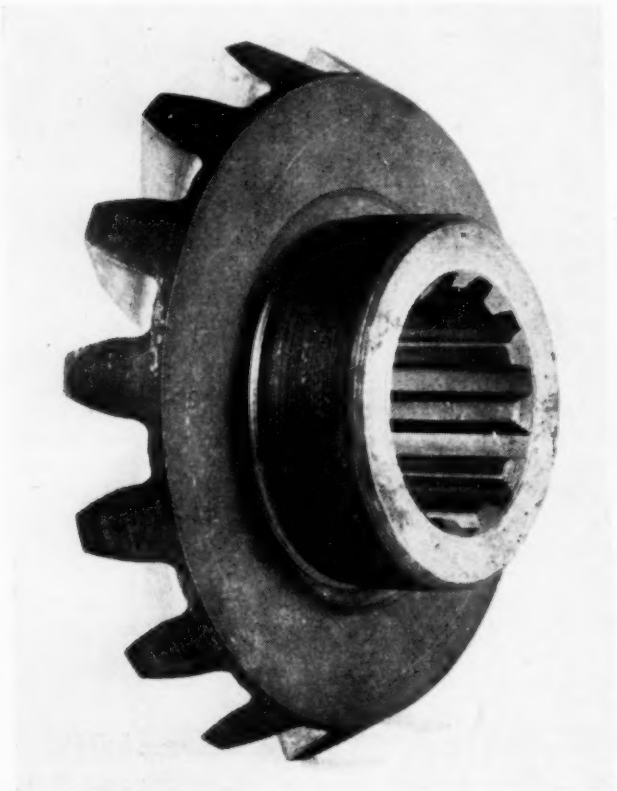
■ Effective Tooth Thickness and Space Width

If a splined shaft has a measured tooth thickness t and its interference error amounts to e, its contour will interfere by as much as e inches with the ideal spline contour



■ Fig. 5 – Redesign of Fig. 4 with involute spline under low-speed gear on sleeve and in clutches for high and second

of the given tooth thickness t. The fit of this spline shaft, or its backlash with a mating member, will be the same as if it had a tooth thickness equal to t plus e. This corrected tooth thickness which expresses the effect of accumulated irregularities on the fit in the assembly is termed "effective tooth thickness." The tooth thickness t as measured is designated as "actual tooth thickness" of the splined shaft. Hence: *The effective tooth thickness equals the actual tooth thickness plus the interference error.*



■ Fig. 6 – Splines under hubs prevent attainment of satisfactory accuracy and functional value

The same reasoning applied to splined holes introduces the terms "effective space width" and "actual space width." Therefore: *The effective space width equals the actual space width minus the interference error.*

The interference error, which is the difference between the effective and actual dimensions of each member, is a measure of the uniformity of the spline and has a definite relation to the wear life of the splined fitting. Excessive interference error causes loss of contact surface and excessive stress in the few contacting spline teeth.

When mating splines are assembled, the fit obtained depends on the effective space width of the splined hole and the effective tooth thickness of the splined shaft. If these two dimensions are equal, a metal-to-metal fit, although only at spots, should be expected even if the actual space width of the splined hole is larger than the actual tooth thickness of the splined shaft.[2] If the effective space width of the splined hole exceeds the effective tooth thickness of the mating member, backlash results, while the reverse condition produces press fit or negative backlash. Consequently, *backlash equals effective space width of splined hole minus effective tooth thickness of splined shaft.*

■ Spline Gages

Various methods of gaging and types of gages are used for checking splines, namely:

Spline ring and master plug gages with full complement of teeth.

Spline plug gages with full complement of teeth.

Snap gages for tooth thickness.

Paddle-type spline plug gages with teeth diametrically opposite.

Tooth verniers, tooth micrometers, vernier calipers, and the like.

Measurements over and between wires.

The indiscriminate use of these various gages has been the cause of much confusion due to differences in the inspection results for which there was usually no consistent interpretation. These differences, which seem to be discrepancies, find a logical explanation if the gages listed above are segregated as shown in Fig. 12.

Gages in the group to the left check the physical width of spline teeth or spaces, specified and identified as the actual tooth thickness and actual space width.

Gages in the group to the right provide a composite check of all elements which contribute to the spline fit. They check the effective tooth thickness and effective space width.

The functional difference between these two groups of gages is fundamental and should never be confused. Inspection results obtained with spline gages having a full

[2] It is conceivable that interference errors on one member are compensated by corresponding clearances on the mating member. However, the probability of such configuration may readily be discounted.

For example, if a set of mating splines with 12 teeth had an identical pattern of spacing errors, it would match in only one out of 12 relative positions. The probability of obtaining perfect matching would be only 0.083 times the probability of simultaneous occurrence of the same spacing pattern or, assuming equal chances for each member, 0.083 times the square of the probability of the independent occurrence of the particular pattern. It will be granted that the resulting probability is slight.

A complete, scientific treatment of these simultaneous pattern occurrences might be valuable in establishing effective tooth thickness and space width on the basis of statistical analyses.

complement of teeth cannot be the same as the inspection results obtained with gages checking actual dimensions. Clear coordination requires that each type of gage be used for its specific purpose, as shown in Fig. 13.

■ Gages for Actual Dimensions

These gages should be designed to check only the actual tooth thickness or actual space width at the pitch line, or any other defined measuring diameter.

Tooth verniers, tooth micrometers, and measurements over or between wires, in general, satisfy this condition. Vernier calipers, which measure involute splines and gears over several teeth by means of parallel contact surfaces give readings which are affected by spacing errors and profile deviations.

Paddle-type spline plug gages should be designed with a definitely high pitch point wherever practical. In other words, the tooth profile or paddle-type plug gages for involute splines should clear the theoretical involute form below the pitch line and be removed above the pitch line by truncation. Normally, these gages have two diametrically opposite pairs of teeth and check only the surfaces of two individual, opposite part teeth clearing adjoining teeth.

Snap gages for checking the actual tooth thickness are usually designed with plain V grooves for pitch line contact. This design ensures a check of the actual tooth thickness at the pitch line only without being affected by profile errors. In order to achieve the same result with serration splines, these snap gages may be truncated and a clearance groove provided.

■ Gages for Effective Dimensions

Although spline gages with a full complement of teeth are held to extremely close tolerances some interference error in the gage itself should be recognized.

In the fitting of spline ring gages to master plugs, the Vinco Corp. found that the actual space width of the average ring gage exceeds the actual tooth thickness of the master plug gage. The difference between these two actual dimensions represents the total amount of interference error in both the ring and the master plug gage. It varies with each set of gages. The variation of interference error, however, is extremely small since the interference error itself is only minute. The following tabulation, based on exhaustive tests on numerous sample gages up to 8-in. spline diameter, shows the maximum and mean interference error between spline ring and master.

Type of Spline	Interference Error Between Spline Ring and Master up to 8 In. Diameter	
	Maximum	Mean
Involute Splines	0.0004	0.0003
Serration Splines	0.0003	0.0002
Straight Sided Splines	0.0003	0.0002

The maximum interference error, as shown, is a limit dimension for the inspection of gages. It represents the greatest difference allowed between the actual space width of a Vinco standard ring gage and the actual tooth thickness of a Vinco standard master plug.

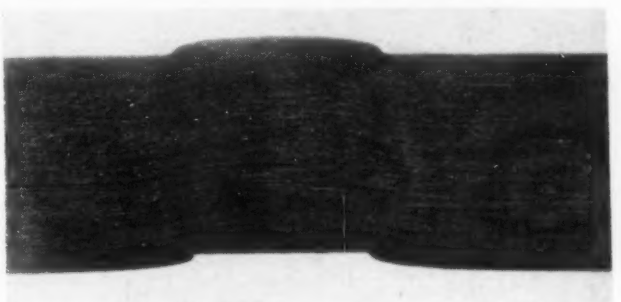
The mean interference error is a gage constant which should be incorporated into the design of each gage. For this purpose, it is necessary to determine the distribution of interference errors between the ring gage and the master plug. Theoretically, it is possible to check all errors of the plug gage and it is probable that less interference error would be found on the plug than on the ring gage. In view of the minute total of interference errors in both gages, however, it does not appear practical to assign a larger interference allowance to the ring gage than to the plug gage. It may be stated, therefore, without deviating appreciably from average conditions, that each spline gage has an interference error amounting to one-half of the mean interference error already tabulated.

Since spline gages with a full complement of teeth are used to check effective tooth thickness and effective space width, they will be designed so that the effective tooth thickness, or effective space width of the gage equals the effective dimension which the gage is to check. In designing spline gages, it will be necessary, therefore, to apply an interference allowance by subtracting the mean interference error of one gage (which is one-half the amount shown in the table) from the effective tooth thickness of the plug gage and be added to the effective space width of the ring gage. From these allowances the actual tooth thickness and the actual space width is obtained for the gages. These are the dimensions checked by means of measuring wires when the gages are inspected.

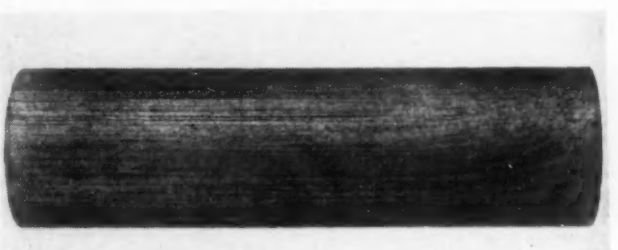
■ Part Specifications

Systematic procedures can be developed from this new treatment of spline interferences. Methods of control, however, require clear identification which is the purpose of the terminology presented above. A revision is, therefore, recommended to the present methods of specifying spline dimensions.

It should be recognized that each splined shaft has an actual tooth thickness and an effective tooth thickness, and that each splined hole has an actual space width and an effective space width. The variations of each of these



■ Fig. 7 – Fiber structure of 1½-in. shaft where offset ⅛ in.



■ Fig. 8 – Fiber structure of 1½-in. shaft where twisted through 45 to 90 deg

dimensions in a production run lead to the existence of the following limit dimensions for tooth thickness and space width:

Splined Shafts	Splined Holes
Maximum actual tooth thickness	Minimum actual space width
Minimum actual tooth thickness	Maximum actual space width
Maximum effective tooth thickness	Minimum effective space width
Minimum effective tooth thickness	Maximum effective space width

To select the proper control dimensions requires a clear understanding of the functional significance of each. A detailed analysis of this significance is, therefore, presented, dealing with splined shafts only. The conclusions reached will apply to splined holes as well.

The *maximum effective tooth thickness* should always be specified. It controls minimum clearance, or maximum interference, and constitutes the most important dimension contributing to proper assembly.

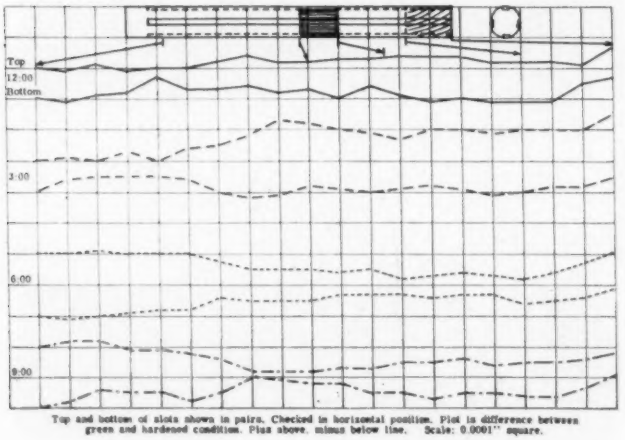
The *maximum actual tooth thickness* equals the maximum effective tooth thickness minus the minimum interference error. It needs no control as long as the maximum effective tooth thickness is not exceeded, since the actual tooth thickness could exceed its maximum value only when the interference error is less than anticipated – which is definitely desirable.

There are, however, two useful applications of a *high limit actual tooth thickness* which is equal to the maximum effective tooth thickness minus the *maximum* interference error. It is not recommended for routine inspection, but for the following purposes:

(a) To check the shaft spline while it is still in the machine.

(b) To determine whether a rejection for oversize effective tooth thickness is due to excessive interference error. If the high limit actual tooth thickness is not exceeded, excessive interference error only can cause oversize effective tooth thickness.

In both cases, the high limit actual tooth thickness aids in establishing a machine setting which will be entirely satisfactory when the maximum interference error is reliably estimated.



Top and bottom of slots shown in pairs. Checked in horizontal position. Plot is difference between green and hardened condition. Plus above, minus below line. Scale: 0.0001" square.

■ Fig. 9 – Chart showing lead checks of four keyways cut in test bar with offset and twist, before and after heat-treatment

Table 1 – Spline Specification Chart

Control System	Shaft-Tooth Thickness Specified: Maximum Effective	Shaft-Tooth Thickness Specified: High Limit Actual[a]	Hole – Space Width Specified: Minimum Effective	Hole – Space Width Specified: Minimum Actual
1. Backlash and Tooth Thickness	●	..	..	●
1A. Refinement of No. 1	●	●	..	●
2. Backlash	●	..	●	..
2A. Refinement of No. 2	●	●	●	..
3. Assembly	●	..	..	..
4. Tooth Thickness	..	●	..	●
Control System	**Minimum Effective**	**Low Limit Actual[b]**	**Maximum Effective**	**Maximum Actual**

[a] High limit actual tooth thickness = Maximum effective tooth thickness – Maximum interference error.

[b] Low limit actual space width = Minimum effective space width + Maximum interference error.

The *minimum effective tooth thickness* equals the minimum actual tooth thickness plus the minimum interference error. It controls the loosest nominal fit with the mating spline, but it does not reflect the possible increase in backlash caused by wear of the profile high spots or deflection of the spline teeth under load. It will be specified:

(a) For spline couplings operating under considerable load, for the use of the minimum effective tooth thickness as an acceptance dimension avoids the risk of rejecting parts with only a few undersize teeth. It will be necessary, however, to assure control of the interference error by careful attention to manufacturing equipment and periodic individual part inspection to maintain the necessary quality of production performance.

(b) For moderate press fits not requiring that the actual tooth thickness of the splined shaft exceed the actual space width of the mating member.

(c) For spline couplings used in mechanisms which require high accuracy and are not subjected to heavy loads.

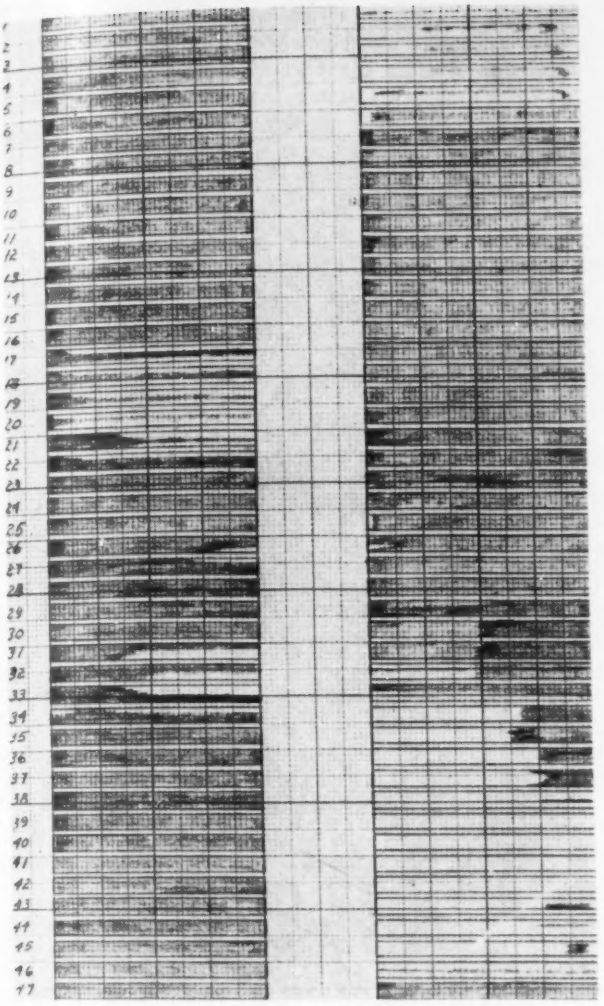
The *minimum actual tooth thickness* should be controlled when it is desired that the backlash will not exceed the absolute maximum even after all high spots of the spline profile are worn off or deflected.

Its control may also be necessary in case of heavy press fit when it is desired that the actual tooth thickness of the shaft exceed the actual space width of the mating member.

It is evident from this analysis that spline data must be coordinated with processing plans and service conditions. The spline specification chart shown as Table 1 is presented to assist in the preparation of spline data consistent with systematic inspection procedures.

Control System No. 1 (backlash and tooth thickness control) assures absorption of interference error within specified limits on tooth thickness or space width; used for spline couplings operating under severe service conditions, and, for heavy press fits requiring that the actual tooth thickness of the splined shaft exceed the actual space width of the splined hole. Care is required in establishing numerical limits, allowing sufficient tolerance for interference error and also for variation in actual tooth thickness or space width.

Control System No. 1A (refinement of No. 1) provides for inspection of work while still held in machine; also, for detection of excessive interference error when effective dimension is found to be beyond the limit established under system No. 1. The high limit actual tooth thickness



■ Fig. 10 – Tooth pattern of hub spline after press fitting showing effect of fiber structure – right: left side; left: right side

(low limit actual space width) facilitates machine setting; dimensional control in routine inspection is otherwise the same as in system No. 1.

Control System No. 2 (backlash control) assures backlash control within specified limits without the risk of rejection of parts with only a few undersize teeth; used where control of interference error is assured by a high standard of production equipment and methods; also, for moderate press fits which do not require that the actual tooth thickness of the splined shaft exceed the actual space width of the splined hole, and where high accuracy is required and loads are light.

Control System No. 2A (refinement of No. 2) combines the features of control system No. 2 with the convenience of a setup dimension as discussed under system No. 1A.

Control System No. 3 (assembly control) assures spline assembly. Tightness controlled, looseness not controlled. It is used where load and accuracy requirements are not exacting, also, for nonreversible, constant loads.

Control System No. 4 (tooth thickness control) provides mere control of tooth thickness without assurance of proper assembly; may be used where selective or individual fitting is desirable, or where backlash allowance is liberal. Tooth thickness control is frequently mistaken for backlash control.

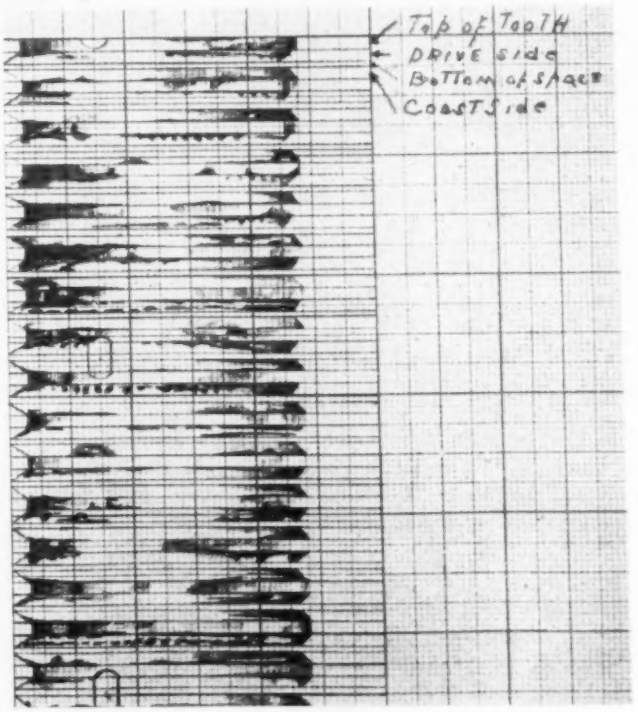
Control systems Nos. 1A and 2A are used by several prominent manufacturers. Control system No. 1, with its refinement No. 1A, constitutes the most rigorous plan attainable for continuous routine inspection of highly stressed splines. Control system No. 2 is extremely popular and is preferred where mere backlash control is adequate. However, the use of this system will permit acceptance of splines entirely within limits of the effective tooth thickness or space width; but with no control of the desired tooth contact. On highly stressed splines, it is imperative, therefore, that control system No. 2 or No. 2A be supplemented by careful attention to the interference error. Much difficulty and confusion can be avoided if these conditions are understood.

■ Interference Allowance

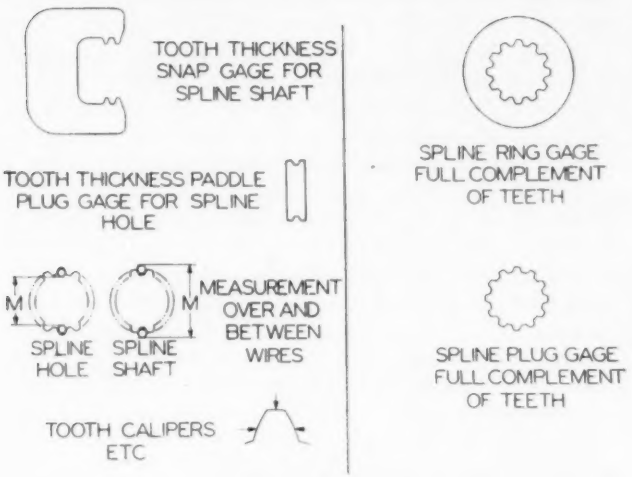
In dealing with spline specifications, it is imperative to distinguish between inspection and design. Each manufactured spline will have some interference error. The designer should anticipate this condition and construct an *interference allowance* which represents the permissible interference error. There are three alternatives in evaluating the interference allowance:

(a) Total absorption of interference error within specified limits on tooth thickness or space width – control system No. 1.

(b) Accumulation of interference errors permissible outside such specified limits – at the expense of actual tooth thickness (increase of actual space width) which is not controlled in system No. 2 and not even in No. 2A, despite the latter's unilateral restriction; the interference error can



■ Fig. 11 – Unwrapped view of press fit spline on end of rear-axle drive pinion showing universal joint action of a supposedly tight fitting


TOOTH THICKNESS SNAP GAGE FOR SPLINE SHAFT
TOOTH THICKNESS PADDLE PLUG GAGE FOR SPLINE HOLE
M
M
MEASUREMENT OVER AND BETWEEN WIRES
SPLINE HOLE
SPLINE SHAFT
TOOTH CALIPERS ETC
SPLINE RING GAGE FULL COMPLEMENT OF TEETH
SPLINE PLUG GAGE FULL COMPLEMENT OF TEETH

■ Fig. 12 – Left: gages for actual dimensions; right: gages for effective dimensions

be restricted, however, by a high production standard; at the expense of backlash which is not controlled in system No. 4.

(c) Partial accumulation of interference errors – control systems Nos. 1, 1A, and 2A when the spread from the maximum effective tooth thickness (or minimum effective space width) to the specified actual dimension is so constructed as not to permit full absorption of all interference errors.

The proper evaluation of the interference allowance depends on reliable estimates of the errors as produced by available production equipment, also on design requirements and on inspection procedures. To formulate a general recommendation for interference allowance values requires examination of many factors and case studies.

It should be stated, however, that some manufacturers determine the interference error on the basis of exhaustive checks of all spline elements on a number of representative parts. Others prefer to have their spline gages fitted to representative production parts; thus, by measuring the actual tooth thickness or space width of both the part and the gage, the interference error is readily established.

■ Scope of Application

The system of actual and effective dimensions presented with this paper applies to splines of every type: involute, straight sided, and serration.

For serration splines, a modification of terminology is necessary because it is conventional practice to express variations of tooth thickness or space width as corresponding variations of pitch diameter.

Pitch diameter, as related to serration splines, is interpreted as a diameter located midway between the major and minor apex diameters.[3]

Since a variation of tooth thickness, or space width, causes a variation of major and minor apex diameters, a pitch diameter can be derived from either the actual or the effective tooth thickness or space width. If the pitch diameter derived from the actual tooth thickness or space width is designated as actual pitch diameter and the pitch diameter derived from the effective tooth thickness or space width is designated as effective pitch diameter, the treatment of specifications for serration splines may readily be incorporated into the general framework of spline fit control.

[3] See p. 249 of 1945 "SAE Handbook."

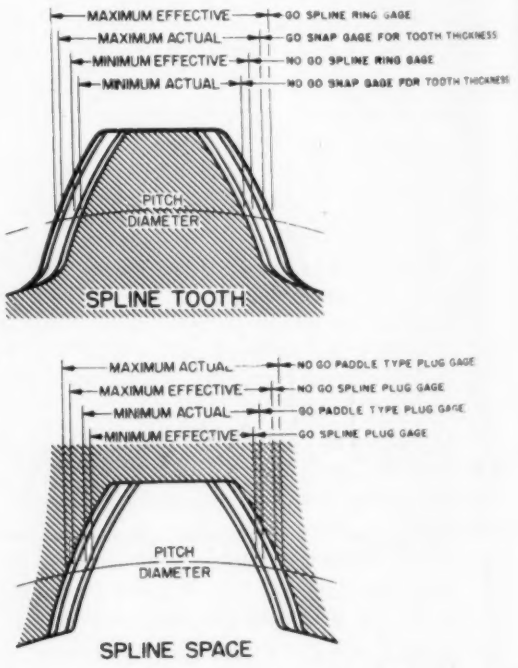
Broaching Applied To New Standards

—H. H. GOTBERG
Colonial Broach Co.

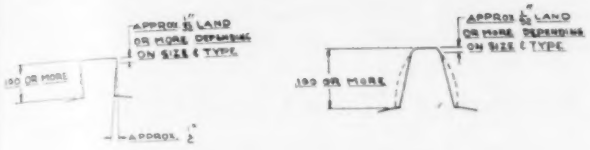
Advanced tool design, together with standardization of cutting tools and of manufacturing facilities, has enabled product engineers to improve the design of many component parts of their products to facilitate tooling, reduce costs, and improve quality. It was natural, therefore, that when the specifications for the new American involute spline standards were established that the tooling factor was considered most thoroughly. As a result, the amount of tooling required as well as the cost of tooling could be held to a minimum.

Many elements have to be considered in the processing of spline shafts and fittings. It is my job, however, to analyze the place of broaching in this field. In this connection I would like to cite one example of how tooling costs can be minimized by adoption of the new spline standards.

First, it should be understood that the cost of an involute spline broach depends primarily on its diameter and overall



■ Fig. 13 – Each type of gage must be used for its specific purpose



■ Fig. 14 – Broach design for internal splines – left: straight side type; right: involute type

length, although other elements, such as the number of splines and the depth of the splines, also enter into the cost of the broach.

As to those primary factors, diameter and length, the length of the broaching tool is determined primarily by the depth of the spline to be cut, the length of the part, and the type of material to be broached. From these, product engineers can usually calculate the torque load and other elements of spline strength. This means, allowing a practical safety factor, that the length of the spline part and the depth of spline can easily be held to a safe minimum, reflected automatically in minimum tooling costs without jeopardizing the strength of the component parts.

Since the depth of the spline exerts a major influence on broach length, the new American involute spline standard was based on a stub tooth form. This tooth form reduces required depth of broach cut, thereby reducing the overall length of broach as compared to the length of broach required for a full depth involute form.

In designing the new standards another factor influencing broach costs has also been taken into consideration. The corners of splines have a tendency to subject a broach to greater wear and breakdown than does the body of the tooth, especially when it is necessary to hold corners to a sharp point. As a result, where relatively close tolerances have to be held on major diameters, difficulty is sometimes encountered in assembling due to the breakdown of spline corners – even though the difference may be only a little more than a few thousandths of an inch. This condition sometimes occurs after even relatively few parts have been broached.

In order to avoid this condition, the splines in the new standards have been provided with adequate radii on the top of the splines, while the component spline shafts are to be chamfered to correspond to these radii. This provision not only ensures facility of assembly of splined parts but also increases broach life materially, since a broach tooth with a radius does not break down as readily as one with sharp corners.

Another important tooling factor in favor of the recommended standards for involute splines is the advantage of reducing costs in the processing of broaches for experimental designs of products. We all know that involute spline broaches are high precision tools and accordingly are rather costly to produce. Usually the cost of such broaches for experimental purposes tends to be prohibitive, especially where only two or three experimental parts are required. From past experience we know that it is almost impossible to locate standard broaches of the exact specifications desired for the experimental purposes in mind. In contrast to this, once these new involute spline standards have been generally adopted, and the broaching tools for them standardized, it will be possible to secure a broach with the exact specifications desired from some other source already using the same tool in production.

Since spline broaching plays such an important part in the manufacturing and assembling of all kinds of component parts, it might also be advantageous briefly to outline here the general method of multiple spline processing.

This procedure involves the following considerations:

1. Broach design.
2. Processing of production parts.
3. Effect of heat generation.
4. Broaching of heat-treated parts.
5. Spiral splining.
6. Gaging.
7. Tolerances.
8. Inspection.

Let us briefly consider these points in detail.

■ Broach Design – Tooth Side Relief

In broaching internal splines of considerable depth, pickup or galling on sides of splines is sometimes experienced. To overcome this, broaches having a spline depth of 0.100 in., or over, are usually relieved on the sides. This is true in either the straight side or involute type of splines. (See Fig. 14.)

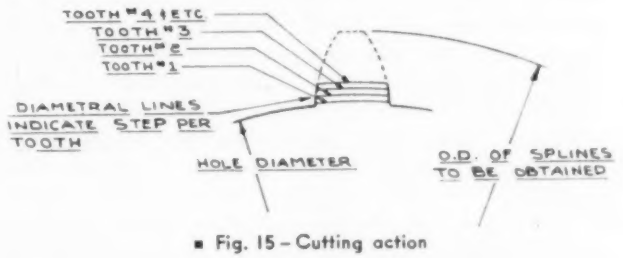
Since the broach cut is generated from the hole diameter outward by diametrical increments of each successive tooth, it is not possible to relieve the sides of the splines on depths less than approximately 0.100 in., since it is difficult to reach the shallow depths with a grinding wheel. However, pickup or galling is usually not experienced on such shallow cuts. (See Fig. 15.)

■ Spacing Errors

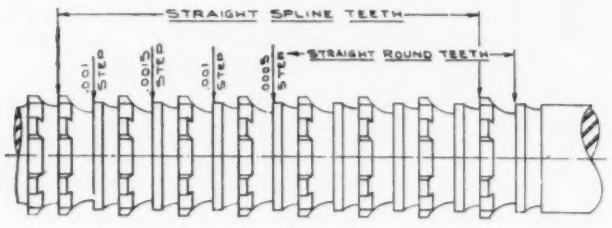
Occasionally, relieving the sides of splines results in a slight drift while pulling the broach through the part. Where extremely close limits are required, especially on involute splines, the side relief is sometimes omitted. Extreme care is necessary in such cases in making the broach. On broaches produced without side relief, a special heat-treat operation can be applied after the broach is completely finished, thereby producing a higher surface hardness on the sides of teeth, which helps to eliminate pickup or galling.

■ Processing Production Parts

In processing parts, the sequence of manufacturing operations is sometimes such that spline arbors are re-



■ Fig. 15 – Cutting action



■ Fig. 16 – Alternate round and spline teeth on finishing end of broach

quired, so that subsequent machining operations can be performed in relation to spline holes. To eliminate the need of spline arbors, a broach design has been developed that incorporates alternate round and spline teeth on the finishing end of the broach. (See Fig. 16.)

This broach maintains a high degree of concentricity between the splines and the inside diameter, even though the limits on the finished parts may be as much as 0.010 to 0.020 in. This type of broach thus facilitates the use of plain round arbors for subsequent machining operations. Where it is important to maintain close dimensional limits on the concentricity of inside and outside diameters of the splines, a broach of this design will assist in meeting the specifications.

Every broach regardless of design or type has a tendency to drift to a certain extent. At times the drift may be very slight, however, and passable parts are produced in the majority of cases. When the limits are held as close as 0.001 in. on both the ID and OD and these limits are of functional importance in the assembly of component parts, it is advisable to recommend alternate tooth design to eliminate the possibility of drifting beyond the required limits.

■ Effect of Heat Generation

We have recently had to produce involute spline broaches to very close tolerances on the chordal tooth thickness, these limits often being held to as little as 0.0005 in. total tooth tolerance. This, of course, requires extreme care in manufacturing the broach, since a slight accumulated error in broach tooth spacing, as well as in tooth width, will infringe upon the permissible limits as to tooth thickness.

Sometimes when a broach is put to work, it will at first produce parts that will permit a "go" gage to enter and "no go" not to enter. However, after broaching several parts, the heat generated in the broach sometimes becomes great enough so that a "no go" gage will enter the parts. It has been thought that the broach could be ground slightly undersize to compensate for expansion due to generated heat, preheating the broach before actually using it, to correct this difficulty. However, in attempting to follow this procedure it will be found almost impossible to maintain production consistency.

Rather, it is recommended that two passes be used where tolerances are sufficiently close to make heat expansion a problem.

As shown in Fig. 17, the first pass broach should leave approximately 0.010 in. to 0.020 in. on each side of the spline. If parts are permitted to cool between the first and second pass, the heat generated in the second pass broach is then not sufficient to result in material heat expansion under these conditions.

■ Broaching of Heat-Treated Parts

When parts have to be heat-treated after broaching, a certain amount of distortion usually takes place. To correct this condition after heat-treat, a sizing broach which is classed as a hard-gear broach may be used. The name derives from the fact that these operations are usually required on gears where the teeth on the outside diameter are hardened to "extremely hard" Rockwell readings. It should be understood, of course, that if the outside gear teeth are to be hardened to more than C-40 Rockwell, the splines must be painted or packed to prevent the splines from exceeding the above hardness. This facilitates re-broaching. Gears hardened to higher Rockwell readings, it is true, have been successfully rebroached but the higher the Rockwell the lower the tool life.

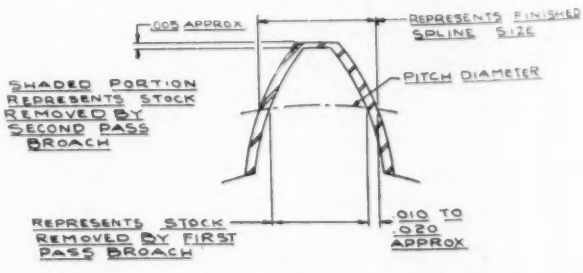
■ Spiral Splining

Spiral splines of both the straight side and the involute type are produced in a manner similar to that employed for conventional straight splines, with the exception that it is necessary to incorporate facilities for generating a helix. In producing a splined spiral shaft, the amount of helix angle which can be developed is not limited, since hobbing or shaping equipment can be geared to produce even very steep helix angles. In broaches, however, a more difficult problem presents itself. In grinding spiral spline broaches, master lead screws are used. The master bar is driven by horizontal motion in the direction of its longitudinal center line. The helix angle, as a result, is somewhat limited since, on excessive angles, the side thrust becomes too great to rotate the bar. However, satisfactory broaches with a helix angle of approximately 60 deg from the horizontal centerline have been produced.

Actual spiral spline broaching is similarly affected. It is necessary to rotate the broach by mechanical means, especially when the helix angle exceeds 15 deg or so. This is usually done by clamping the part rigidly to a face plate on the machine and rotating the broach by means of a helical drive head actuated by a master spiral lead bar.

■ Spline Plug Gages

Spline "go" gages are usually designed to allow for clearance on the outside diameter as well as on the hole


.005 APPROX
REPRESENTS FINISHED SPLINE SIZE
PITCH DIAMETER
SHADED PORTION REPRESENTS STOCK REMOVED BY SECOND PASS BROACH
REPRESENTS STOCK REMOVED BY FIRST PASS BROACH
.010 TO .020 APPROX

■ Fig. 17 – Two passes should be used where tolerances are sufficiently close to make heat expansion a problem

diameter, since the primary purpose is to check the width of the spline and for possible spacing errors.

The spline width on the "go" gage is usually held to the low limit of part specifications with a minus 0.0001 tolerance. (See Fig. 18.) "No go" gages are likewise designed to allow for clearance on OD and ID, with the spline width held to the high limit of part print specifications, and a tolerance of plus 0.0001 in. It has been found that holding to these tolerances will allow for maximum tool life on broaching tools. These specifications apply to gages for both straight side and involute straight splines as well as to spirally broached splines. The length of spline plug section should be equal to the length of the part as nearly as possible, to ensure that the splined hole will be straight through the entire length of the part.

■ Spline Ring Gages

Spline ring gages for checking straight side or involute type of splines differ slightly in design.

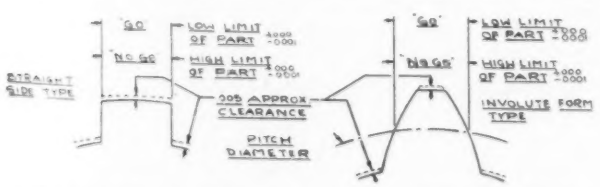
Spline ring gages of the straight side type are of open end construction and can be made to check the OD of the shaft as well as spline width and spacing error. (See Fig. 19.)

The involute type of gage must be either broached or shaped and lapped or ground, depending upon the equipment available, as well as the sizes under consideration. The involute type is designed with clearance for OD and ID of the shaft. However, tolerances on the spline width and chordal thickness are the same in either type, as follows: "go" gage spline width held to high limit of shaft with a plus 0.0001 tolerance. "No go" gage spline width held to low limit of spline shaft with a minus 0.0001 tolerance. You will note that the gaging tolerances as outlined will allow maximum tool life. However, difficulty is sometimes encountered in determining in which direction gaging tolerances should be allowed. That is, whether the manufacturing tolerances of a spline "no go" plug gage should be held to the high limits of a part with a plus tolerance or with a minus tolerance; and whether a "go" spline plug gage should be held to the low limits of the part with a minus tolerance or a plus tolerance. Since there is no standard rule as yet governing these conditions, it is almost impossible to maintain consistency of gage design. We hope that soon such gages also will be considered for standardization.

■ Tolerances

We have found, in rare instances, that where very close tolerances on spline width are specified, the customer has actually withheld the total tolerance and specified less tolerance than really allowable in order to be sure that the tools would be well within the accuracy limits required. This procedure does more harm than good, since when an extremely close tolerance is given, the designer as well as the broach grinder is instructed to hold the spline width as close to the high limit as possible so that the tool will not cut undersize.

When a tolerance of 0.0005 is actually permissible and instructions are issued that a 0.0003 tolerance is to be held, the chances are that the broach will tend to cut oversize, whereas if the entire 0.0005 tolerance had been submitted, enough leeway could have been given to assure holding to the correct tolerance as given on the part print specifications.



■ Fig. 18 – Spline width tolerance on "go" and "no go" gages

■ Inspection

Inspection gages frequently differ from gages used in actual production departments. That is, production gages are held to closer tolerances than inspection gages to allow salvaging of parts not passable by production gages. When this procedure is followed, it is suggested that actual dimensions be submitted to the gage manufacturer rather than master gages "to be copied," since the master gages may vary dimensionally from time to time, making reproduction difficult.

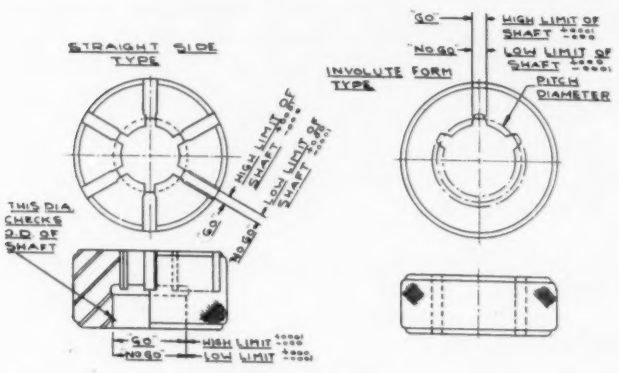
Use of Single Radius On Broaches Analyzed

—S. O. BJORNBERG
Illinois Tool Works

The thought has been introduced that the profile of the internal fittings in our involute spline system might be more advantageously constructed as a single radius instead of following true involute geometry. Advantages to the broachmaker and the spline user are claimed for this principle.

In order to arrive at some basis for appraisal of these claims we have attempted to analyze the effect of applying the single radius to splines of various pitches and various number of teeth. This was done by superimposing, on a magnified layout of a given spline profile, a radius of such size and at such position as will follow the involute contour as closely as possible. We found that, in the finer pitches and with a large number of splines, a single radius can be made to correspond with the involute with less error than we can measure. On the other hand, in coarse pitches and with a small number of splines the discrepancy between the radius and the involute becomes very pronounced. When found at the major diameter the error is in the direction of interference, and when found at the minor diameter, it creates clearance between the mating profiles.

By adjusting the value and position of the radius, the errors of interference can be made equal to the errors of clearance and, in this way, the total error is held to a minimum. Under these conditions the contact is theoretically concentrated at the point of interference, a condition which is undesirable in principle. For better conditions of contact a radius can be established that eliminates the interference, but this increases the amount of clearance and therefore increases the total amount of profile error.



■ Fig. 19 – Open end construction of spline ring gage of straight side type

Thus attacked, the problem becomes a matter of how much error might be permissible in the involute profile, and raises the question as to whether the standard of accuracy applied to a radius profile should be any different than that applied to an involute profile.

Seeking a quantitative basis of comparison between the two types of profiles, we followed the principle of a single standard to be applied to both. For the sake of discussion we used, for all pitches, a figure of 0.0005 in. for the maximum allowable deviation from true involute.

Using this figure we established the minimum number of splines in the various pitches which would permit a radius profile to measure within the allowable profile error.

We found that the radius profile could be used for all the numbers of teeth in the standard in pitches of 24/48 and finer. In 20/40 pitch only the 6-tooth spline showed errors of more than the allowable 0.0005 between the radius and the involute. In the other pitches the minimum number of teeth for which the single radius can be used are:

9 teeth for 16/32 pitch
10 teeth for 12/24 pitch
11 teeth for 10/20 and 8/16 pitches
12 teeth for 6/12 and 5/10 pitches
13 teeth for 4/8 and 3/6 pitches
14 teeth for 2.5/5 and 1/2 pitches

On the basis of a total error of 0.0005, then, a single radius can, theoretically, be employed on the profiles of many of the splines without inviting any criticism of profile accuracy. In practice, however, full advantage cannot be taken of this range because of errors of contour and positioning of the radius being applied. Where these errors add up to equal the total allowable profile error, their direction must be controlled or there will remain no possibility of a radius profile meeting the standard of accuracy for the involute.

Here, then, is applied that principle of dimensional engineering which deducts from the manufacturing tolerance any error incorporated in design. Where a radius is to be substituted for an involute, the maximum free tolerance permitted in manufacturing is that remaining after the basic error between radius and involute is subtracted from the original profile tolerance. The broachmaker, in choosing to work with a radius, then, must do so with the thought that he can more easily meet the closer tolerance required on the radius than the tolerance on the original involute.

To dress either an involute or a radius on a grinding wheel requires a special dressing fixture and, where both types of fixtures are available, it is to be doubted whether a radius can be placed on a broach profile with any higher degree of accuracy than can an involute. Where no concession is granted to the radius profile, therefore, it would seem that the handicap of operating with reduced tolerances should make much less attractive the apparent ease with which a radius can be substituted for an involute.

If it were found that a radius profile could be permitted greater deviation from the true involute than a profile of basic involute construction, a range of splines would be developed within which the radius may be employed with the same manufacturing tolerance as for an involute. Within this range, then, the process of dressing a radius could compete on equal terms with the dressing of an involute.

The only defendable direction for such a concession to radius construction is that which would create clearance between the mating profiles near the minor diameter. Clearance at this point would sometimes be of benefit in the assembly of the spline members as the tips of the teeth in the internal fitting frequently cause interference troubles.

Clearance at any point, however, essentially defines less area for contact, a condition affecting the basic concept of the involute system for splines. As Mr. McCain has related, the application of an involute to spline profiles was inaugurated largely because the self-centering characteristics of the involute and the greater accuracy with which it can be produced give a greater area of contact than is ordinarily found in other types of splines. It is not surprising, therefore, to find much opinion opposing the arbitrary destruction of a portion of this contact as a concession to design preference on the part of the broachmaker, or poor workmanship or rough handling on the part of the spline manufacturer.

Manufacturing Problems Said To Hinder Use of New Spline

—B. F. BREGI
National Broach & Machine Co.

We believe that the paper by Mr. McCain represents a good job of engineering work; however, it leaves some things to be desired from the manufacturing standpoint.

The shaft splines as presented can only be processed by ordinary cutting methods, such as hobbing, gear shaping, and possibly milling. If further refinement is necessary, as is often the case, these splines must be ground. This is costly and often unsatisfactory.

Shaving and/or lapping cannot be employed as the standard is now set up. It must be remembered that many manufacturers, including large auto and transmission builders, do economically shave and lap and will probably continue to do so because of accuracy, finish, and cost. This may mean that the proposed spline cannot be used.

We need modifications or another system of "involute splines for modern production." We believe that it is possible to do this by using:

(a) Standard cutting tools, such as hobs already available.

(b) Shaving and/or lapping, which is already a must in many plants because it is modern, fast, accurate, and economical.

(c) Shaving and/or lapping can be used to reduce the interference error, thereby reducing the difference between actual and effective dimensions.

(d) Standard inspection tools and methods can be used, much of which are already available in many plants, at least for the shaft part.

Complete data concerning such a spline standard will soon be available for those interested.

Larger Corner Radius Brings Improvements

— RONALD V. HUTCHINSON
General Motors Corp.

Seemingly Fig. 11 (of the paper by Mr. McCain) shows the two basic rack profiles acceptable to Mr. McCain and associates.

Topping hobs with the full rounded tooth enable centering contact to occur on the shaft major diameter, given other adequate clearances.

This allows:

(a) Indicating tops of shaft splines for true-running inspection.

(b) Adequate room for shot-peening the root fillets.

(c) Potentially improved smoothness of root fillet as generated.

The flat topped hob illustrated, capable of generating substantially a cylindrical root, allows, as mentioned, centering on the minor diameter, again given adequate clearances.

But the small corner radius tends to generate root fillets small and fairly hard to reach by shot-peening, that is, in the finer pitches.

The roughness of the generated fillet increases somewhat as:

(a) Number of splines decreases.

(b) Corner radius of hob tooth decreases.

(c) Offset of center of corner radius from generating pitch plane increases.

All are for conditions otherwise constant, assuming truly mounted and accurately sharpened hobs.

Because of the proved increase in life of parts consequent to proper application of shot-peening, serious consideration should be given the "shot-peen-ability" of work produced with flat topped hobs.

It seems the proposed corner radius 0.075 x DP could be advantageously increased from the dual viewpoint of easier shot-peening and a little more hob durability between successive sharpenings.

■ Bjornberg Discussion

Mr. Bjornberg discussed circular arc versus involute broaches for provision of spline broach teeth.

In the proposed new splines, the pressure angle is great and the tooth depth relatively shallow, so the *change* in curvature along the used portion of the involute is not great.

Three arbitrarily chosen points anywhere along the involute of the flank may be connected successively by a pair of chords, whose perpendicular bisectors in turn intersect at the center of a circle upon which the three points uniquely lie, so the chords are common to both involute and approximating circle. Divergence of circular from involute arc is controlled merely by choice of point positions, and absolute coincidence ideally occurs solely in the three points. Location of center of the approximating circle, and its radius, are simply, although somewhat tediously determinable, enabling the basic data upon which to build up wheel dresser design.

Until one makes relatively enormous teeth, shape divergence need be of very little moment.

For many applications, one may go somewhat further than Mr. Bjornberg and substitute flat surfaces tangent to the involute, for either involute or circular arc shapes produced by the broaches.

Let this tangency be taken near the half depth of the tooth—profile clearances at both major and minor diameters then can result. Some folks like this.

Contact under zero torque is then ideally line contact, and changes to strip contact as the surfaces deflect elastically.

When the hub is appreciably softer than the shaft, the first high torque application can "set" the shaft into the hub and establish excellent contact.

This notion may be incorporated in the manufacturing scheme, particularly when the parts transmit torque in only one direction.

So, while establishment of standards for splines, splined holes, tools, and gages definitely is a worthy cause, we should still be willing to examine it critically.

Excessive Index Errors Affect Spline Capacity

— FOREST R. McFARLAND
Packard Motor Car Co.

We have had a number of experiences in the operation of splined fittings on which we have had to emphasize the importance of accuracy of index and proper alignment of splines.

On the Rolls-Royce Merlin aircraft engine, produced by Packard Motor Car Co. during the war, we had a problem of fretting and breakage on the driving shaft between the crankshaft and the reduction gear pinion which occurred in the test running of these engines. While service trouble was not experienced, corrective steps were indicated. This shaft was 6.3 in. long with a 2.5 in. PD involute spline 1.3 in. long at the front, and a 4.5 in. PD spline 0.5 in. long at the rear. Under load, the front end centered itself in the pinion and the rear end centered itself in the crankshaft which traveled in an orbit equal

to the front main bearing clearance, as evidenced by the load diagram and marking in the bearing.

The front spline showed very little fretting, whereas the rear spline pitted consistently. The difference was apparently due to the fact that the front spline locked itself in the pinion due to the load being taken off at the smaller radius. This resulted in a higher tangential load which caused deflection of the front portion of the pinion to accommodate the movement of the torsion shaft. The rear spline apparently slipped axially due to the fore and aft movement resulting from the "whirl" of the rear end of the shaft, since the load at the rear was taken off at a greater diameter, which resulted in lower total loading. To this movement was added whatever movement was present due to crankshaft flexure caused by the explosion loading of the first cylinders. Had this program been continued, locking of the spline by a higher pressure angle on the teeth, together with introduction of flexibility in the flange of the torsion shaft, would, no doubt, have been considered to improve this condition.

This seems to lead to a principle in design that it is desirable, in spline fittings, either to lower the spline loading to where it can be "lived with" or raise the loading (not necessarily the unit pressure) on the spline to lock it, and take the misalignment present by introducing proper flexibility in the mating parts if possible.

A very few failures of the front splines of the same shaft occurred under test. The first cases were due to index errors. The latter cases resulted from excessive loading of the splines at the rear end of the teeth, occasioned by torsional deflection of the shaft. The dimensioning of the splines was changed so that the sides of the teeth were from parallel to 0.0005 light at the rear end. Splines run with the relief at the rear showed definitely better bearing after model test.

The effect of index errors was further shown by a comparative test of three designs of splines in a supercharger drive shaft. All the tooth dimensions of the splines were the same. The external spline in one case was flush with the internal spline at the driving end. In the second case the external spline protruded ⅜ in. In the third spline the external spline ran out to a diameter the size of the major diameter.

These three types of splines were run against each other in a testing fixture which oscillated them, two at a time, through the same amplitudes. The failures originating at the root of the spline were not too consistent. Finally, no failures occurred, although the stress range of the splines was increased until the top limit was at, or initially above, the yield point of the material in the round-shaft section. At this time the index of the soft hubs was checked and found to be definitely better than at the start of the test. This was apparently due to the correcting influence of the external splines which were definitely harder, and which had much less index errors than the internal splines.

From this experience we concluded that excessive index errors can have as much effect on the capacity of splines as a difference in the design of the splines which should obviously increase the capacity. This experience has already been described in detail.[4]

[4] See Proceedings of Society for Experimental Stress Analysis, Vol. 3, No. 1, 1945, pp. 112-121: "Experiences with Highly Stressed Aircraft-Engine Parts," by F. R. McFarland.

The question of using a radial dresser to approximate an involute curve depends entirely upon the amount of calculated error shown to be present. We used this type of dresser satisfactorily on a supercharger driving shaft spline. This spline was of fine pitch and had a comparatively large number of teeth so that the theoretical error was very small. If the calculated error were less than 0.0001 between the pitch radius and the tip or the root, I believe this method would be satisfactory.

With regard to the gaging of heavily loaded splines, we have stressed attempting to check index errors rather than the thickness of the teeth, except in the case of supercharger impeller splines where a very close fit on the teeth has been considered important in order that unbalance will not creep into the assembly due to repositioning of the impellers on the shaft after the assembly has been initially corrected for static and dynamic balance.

It appears difficult to gage splines in a simple way and maintain index errors without also holding the thickness errors of the teeth to closer limits than is usually necessary except where close fits are required. We have spot checked ground splines for index errors only, to be sure that the processing was correct, on accurate dividing heads or spacing checkers rather than depend upon gages for our index check. The tapered type of gages, for internal splines employing two teeth for size and all of the teeth to establish index errors, the difference in standout, representing index errors, appears to be one method that has merit for segregating index errors from tooth space errors.

MOTOR TRUCKS

continued from page 302

trucks, but the engineers in the plastic industry have a fruitful field of study with respect to the needs for the truck operator's environment. Weight-saving possibilities, of course, would also be expected in connection with all these desirable features.

■ Light-Weight Metals

The war has occasioned a great expansion in facilities as well as research in the light-weight metal field and, in the next few years, the light-weight industry will force itself more and more into the truck picture. There is, however, need for a critical appraisal of the necessity for such materials in trucks, and the basic cost which must be met if extensive production use is to be possible. It has long been in the premium class and in the years to come will find ready acceptance if we get around to building vehicles specifically for industry applications.

The run-of-the-mill truck users will not consciously pay any premium for light-weight metal until it can be readily demonstrated that some economic gain is possible.

The fabrication and maintenance problems are minor compared to the increase in gross cost per vehicle.

It is hoped that some of the things presented will be the subject of real analysis and study that will bring more clearly to the fore the need for proper appreciation of our truck transportation problems.

AIRCRAFT Approach to Automobile Body Design

by MAC SHORT
Lockheed Aircraft Corp.
and W. E. MILLER
Consulting Engineer

THERE frequently comes to the attention of an engineer in one division of our automotive industry, the doings or the developments in other divisions that have the appearance of being especially interesting to him. Therefore one wonders whether or not there may be some aspect of his associates' work which would be helpful or enlightening to his particular studies or assignment, either immediate or future. We have gone through this reasoning many times, in an effort to determine and evaluate the excellent engineering and production accomplishments of the motor-car industry and the body activity in particular, as they might apply to the aircraft industry. Now the tables are turned – the passenger-car body engineers have posed the question – how would the aircraft engineer approach the problem of automobile body designing?

THE fundamental difference between land and air vehicles, the authors explain, is that the plane is supported in the air dynamically, while the car is supported on the ground statically, thus accounting for the importance of weight, strength, and shape in plane design, but not necessarily so in car design.

The authors feel that the aircraft approach may not be directly applicable to car body designing, but they point out that the same basic principles that make for efficient weight, structure, and performance of airplanes can and should unquestionably be useful tools of the automobile body designer.

THE AUTHORS: MAC SHORT (M '25), vice-president of Lockheed Aircraft Corp. and 1943 president of the SAE, is recognized as one of this country's outstanding aeronautical engineers. He began his aviation career in the U. S. Army Air Service during World War I, and subsequently graduated from Kansas State Agricultural College. He later became an aeronautical engineering instructor at M. I. T., and was instrumental in founding both Stearman Aircraft Co. of Wichita, and Vega Aircraft Corp. W. E. MILLER (M '26), who for nearly four years was in the employ of Vega and Lockheed Aircraft Corps. on various projects, has resumed consulting practice, serving custom automobile and commercial body builders. He received a broad background of experience through association with a number of custom automobile builders, such as Robert Thompson Co., and Endurance Steam Car. In 1928 he worked in General Motors Art & Color Section, and from 1929 to 1933 he was a body stylist at Packard. From then until he joined Lockheed, he was a consulting engineer.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 8, 1946.]

The authors are aware of the controversial nature of such a subject. However, it has been agreed briefly to outline, discuss, and illustrate certain of the elements of an aircraft approach to the question. Let us state here and now that this material has been prepared as a somewhat academic study reflecting solely our personal views, with no intention of experting the excellent work and records of the passenger-car engineer and his body specialist. It is to avoid the old bromide about an expert away from home and also the oft-quoted remark of W. B. Stout: "If you accept their assumptions, each speaker is right," that this statement is respectfully and earnestly submitted.

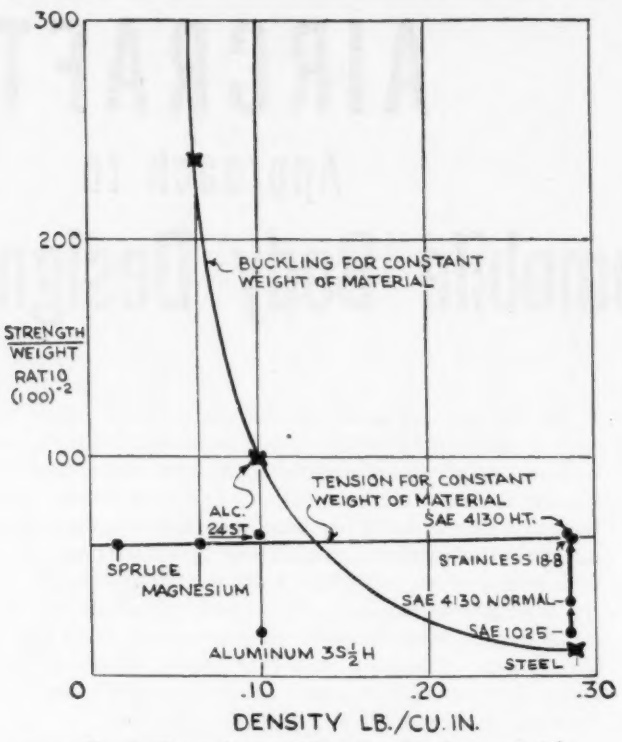
It is feared that you shall question the soundness of the following study, for we have found it difficult to develop statements that make our approach readily useful to a group of engineers, designers, and stylists who have been successful in keeping the American public so pleasantly dissatisfied.

■ Basis of Approach

The material presented herewith is not a word description of a dream-car body or anything remotely resembling it. Instead, the paper is an attempt to point out to the passenger-car body engineer some of the information, some of the methods, and some of the philosophy in common use on the aircraft side of this general automotive engineering field. You need not be one of our particular brand of engineer to make an aircraft approach, but simply to know what has been done in this special field of engineering. It's not a state of mind, it's a process. This being the case, then what is the approach?

This question seems best answered by stating a fundamental difference between an air and a ground vehicle. It boils down simply to this – the airplane supports itself by dynamic reaction in the air, while the motor car is supported by a static reaction on the ground. Maybe it could be said another way – the plane must always move through the air fast enough to support its weight or it will fall out of the sky, while a car just doesn't need this same treatment to keep from falling off the road. Herein lies the major difference, the principle that makes *weight, strength, shape,* and a host of other parameters so basically acute – so vital – in plane design, but not necessarily so in car body design.

To say that one could or should apply these basic parameters to body engineering, with the same need as they are applied to plane design, and expect to come up with a body design that makes sense to the management and public, is entirely too presumptuous. What may be reasonable, however, is to keep in mind the approach that the aircraft



■ Fig. 1 – Strength to weight ratio of various materials

engineer makes toward his design and consider what might be done with this approach when applied to some of the major elements in automobile body design.

■ Weight and Materials

Materials bear an umbilical connection to weight, thus they will be discussed jointly. It is encouraging to observe the general interest in light materials and principally light metals, for all ground vehicles, and doubly so in the passenger-car field.

Probably there is no factor so characteristically aircraft as weight control and light materials. Weight saving or a good relationship between empty and loaded weight is not a fetish of ours, it's the very foundation of economical transportation – land or air. With this sweeping statement you are probably prepared to hear us indulge in a relatively restrained flight of imagination, introducing every sort of material and proposing all sorts of fly-weight figures. This is furthest from the truth, at least in the aircraft atmosphere in which we have been living these past war years. We have had to make our "fantasias" work, too! But it does seem that a realization of the fallacy, that in weight lies strength, is in order.

Our approach is naturally prejudiced toward light metal and generally the aluminum alloys. However, as for the body shell, until the car could be sold to the customer as a plane or a bus is sold, that is, on a long-term cost of transportation basis, it would appear that any approach to the use of aluminum alloy bodies will be definitely circumscribed – yes, ruled out entirely. There does seem to be a great deal of logic to the use of aluminum sheets and die castings for the secondary body parts, hardware, and miscellaneous sheet metal parts around a body. The engine hood (or bonnet to you) and the baggage compartment lid are good examples for the employment of aluminum where weight savings pay off. The effort required to lift these surfaces would be more acceptable to all concerned, while permitting a reduction in weight of hardware, springs, and so on. These shapes can be drawn in dies, assembled simply, and fortunately are not being dinged-up during normal usage of the car on the city streets. Another item that seems worthy of a look is the front and rear bumper assemblies. If there could be a break in the fad for glittering bumpers, then possibly the 70- to 100-lb set of chrome-plated steel units could be replaced by, say, formed dural sections, color anodized, for possibly half the weight of this part of the body assembly.

In passing, it should be stated that to rush in and simply introduce a light-weight material or a high-strength, thin-gage steel to an existing structure, and by so doing assume to have followed aircraft practice is furthest from the truth. Metals and other materials successfully used in the plane and which permit us to keep the weight down must be carefully surveyed before selection and fabrication for the ground vehicle. Fig. 1 is a convenient chart to remember; it illustrates the strength-weight ratio of the common materials plotted against density: first for tension, then for buckling.

You have all seen 24S-T sheet fabricated in plants during the war, where those seemingly flimsy airplane parts gave the production people no end of trouble. There is no question but that the processing of this particular material is out of our discussion, but there are aluminum and even magnesium alloys that have fabricating possibilities.

There is now being developed industrially a new common alloy, known as Reynolds Metals Co. No. R311, with tensile strength properties intermediate between those of 3S and 52S. The new alloy is just being placed in production. The formability of R311 in its various tempers will closely approach the formability of 3S in its respective tempers. Resistance to corrosion should be at least as good, if not better, than that of 3S. Doubtless Alcoa has a similar alloy coming out for the "imagineering" postwar needs.

Whether or not light metals could be successfully employed in body construction nearly as much as in aircraft, only the factors of development and adaptability toward the reduction of weight will foretell. It depends on how long you wheel around about 3000 lb of materials to carry one or two people at an average of 40 mph. As for the plastics, the lightness, strength, quietness, and insulating properties make their consideration in aircraft extremely interesting. It is ventured you know more about them already than do we. One thing is sure though, no all-plastic roof – it's out.

■ Structures and Analysis

Since studying this phase of the subject, collateral reading of earlier and quite recent papers by passenger-car body and frame engineers has brought out the apparent perennial discussion: whether there is to be or not to be a frame and a body or a "unitized" structure, as some authors describe the assembly.

It would, therefore, be well at this juncture to state the interpretation placed on the body assembly. It shall be assumed that the frame, regardless of its extent or what it might be called, acts with the sheet metal body shell as a structural unit. For that matter one should also consider the fenders, the hood, the baggage compartment lid, and

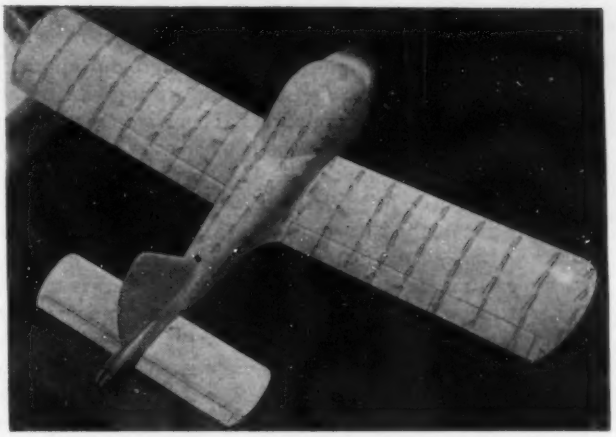
possibly the bumpers and under pans of sheet metal, that is, all members capable of carrying some over-the-road load. This seems like a reasonable approach in an effort to get as much strength as possible from these metal parts. Here seems to be one technical approach of ours which would find use in auto body engineering, provided you subscribe to the functional philosophy.

The possibilities of aircraft structures applied to motor trucks and buses have already been discussed by one of the authors.[1] At the risk of repetition I should like to restate a few points brought out in that paper, believing they apply equally well to passenger-car body as to truck and bus body design. I quote: "It may be helpful to state the philosophy of the aircraft structures engineer in approaching his problem. It can be expressed simply in two objectives: first, to make the best possible use of all the structural material at hand, and secondly, to make the structure direct and continuous. . . . If each part of the structure carries its maximum load and discontinuity is avoided and no low stress intensity allowed, then the structure is assumed to be working at its best strength/weight ratio. If the reverse is the case, then the pieces or parts of the structure should be eliminated or redesigned. . . ." To be sure the aircraft designer cannot always rigorously adhere to this philosophy when developing his own vehicle. He would probably have a much more difficult time making his point with a car body, as probably in the car, comfort and accessibility come before structural efficiency.

It is our firm conviction that the development of the body lines, that is, the lines that establish the shapeliness and strength, can be identical and the structure need not be separated. Surprisingly enough, where the same materials carry out the dual functions of structures and shape of a body, a favorable strength/weight ratio develops. This is what has happened in airplane bodies or fuselages, and one would be inclined to predict that the passenger-car body will undergo the same evolution. Surely a heavy chassis structure with an equally heavy shell perched on top is not the most efficient use of materials or structures. However, if the economics of standardized body application to various chassis models continues, which, mind you, may be logical, the evolution will be much slower.

Whether the design calls for all-monocoque body with no frame or for some of each, a rigorous stress analysis of the *basic loads* seems desirable if not necessary. One may not be able to take full advantage on the drafting board of the research and theoretical data obtained from the laboratory, but the less arbitrary or empirical these basic loads are, the better the body structure will be able to carry the applied loads for a low total weight. These loads form the backbone of the stress analysis, the static or dynamic test, and later the operational stress measurements. It is appreciated that to determine all the probable motor-car loading conditions and combinations thereof requires a tireless investigation. However, an envelop of basic loads seems reasonable, and as these are principally ground reaction loads with practically no important air loads, the structural requirements that must be satisfied could be fairly rationally determined. A detailed analysis and static test could be justified in our approach and a budget of many thousands of hours would not be uncommon for this work. Details, such as spring attachments, engine mounting points, and other strong points, distribution of seat loads, steering loads, and all highly concentrated loads, would be investigated.

[1] See SAE Transactions, Vol. 53, January, 1945, pp. 1-9: "Possibilities of Aircraft Structures in Ground Vehicles," by M. Short.



■ Fig. 2 – Model of small passenger plane mounted in wind tunnel and undergoing flow investigation

In selecting a material for a particular application the aircraft engineer is usually concerned with the ultimate strength. When designing to the ultimate strength of metals, a factor of safety equal to 1.5 is used. It might rightfully be dubbed a factor of ignorance. The structure is designed so that the material is stressed to the yield point. For all materials in common usage, this coincides very closely with the ultimate strength divided by the prescribed factor of safety. For example, SAE 1025 steel has an ultimate strength of 55,000 psi, which, when divided by our 1.5 factor of safety, gives a maximum working stress of 36,000 psi, and which coincides with the yield strength of the material. This is also true in the commonly used Alcoa 24S-T aluminum alloy in which the ultimate strength is around 64,000 psi and the yield strength is 42,000 psi.

With reference to light-weight ground vehicles, it has been noted that working stresses under 1 g loads are of the order of 1600 to 2900 psi in tension and from 1700 to 3100 psi in compression. Using an acceleration of 4 g the average tensile stress becomes 9000 psi, which is about one-quarter of the allowable yield strength of our 24S-T aluminum alloy. In an airframe designed for the same acceleration the material would be worked to the yield strength.

Because of fatigue, motor-car body and frame designers may object to our general design practice of working materials up to high stresses. The airframe stress engineer is not unmindful of the fatigue problem, but in determining the structure and checking its strength he is inclined to consider fatigue *not* as a primary factor but as a secondary one to be eliminated after structural proportions have been established. Local fatigue failures frequently occur, due to violation of the essential rule of continuity of structure. Examination of the structure for local stress-raisers and designing them out rather than the "beefing up" to stop fatigue failure is our approach.

But then, even with the blessing of the boss and the budget to get weight down, the type of detail structural analysis mentioned may not result in sharp weight savings for the body assembly. You may have anticipated from all this analytical and test work a higher percentage; however, the results should form a starting point from which further intelligent weight reduction could continue.

■ Shapes and Lines

In this division of engineering and design, our approach would run head-on into the styling or art approach rather than the technical or analytical. There seems to be an unusual amount of folklore about what constitutes streamline or aerodynamic lines.

In the first part of this paper, the underlying difference between the method of support of a plane and a car was stated. Although the motor car does run its entire existence on the ground, and for this we have considered it to be a "static reaction" vehicle, the entire body surface operates in and through the same air as the plane. It travels in and through a fluid having mass and viscosity, which we know produces forces on the body in motion. Bernoulli's theorem still holds, hence these forces affect performance, power required, and the handling qualities of the car and so bear consideration.

The scope of aerodynamics as applied to the design of a motor car may be limited entirely to the streamlining of shapes for the purpose of reduction of drag – the force tending to retard forward motion of the vehicle. Other components of force, although important to roadability of the machine, particularly at high speeds, are secondary in magnitude. The industrial stylists, together with education of the public by the sales departments, have done a very creditable compromise job of streamlining. We must recognize, in considering *any* approach, that it is necessary to cater to the artistic rather than the purely technical sense in establishing passenger-car body lines.

It will be pointed out shortly that the power required to overcome the air drag of the modern automobile traveling at normal speeds (60 mph) is relatively quite low. In fact, it is quite likely that acceleration and weight are the determining factors for maximum power requirements. Therefore, the following remarks are based on the assumption that the volume of motor-car sales may justify some aerodynamic investigation of the body shapes and lines, even though the net gains in performance and economy may be small.

The total drag force on a body in motion, for convenience of visualization, is divided into three parts, which are termed induced, profile, and parasite drag. The induced drag is that caused by the basic body's deflecting the air stream and the resulting change in momentum of the air. In the case of the motor car, no aerodynamic support of the body is required, and the induced drag may be entirely eliminated, although care should be taken to avoid any abrupt change in the resultant direction of flow in order to keep the induced drag to a minimum.

The profile or form drag is that remaining after the induced drag has been removed from the total drag of the basic body. The profile drag is caused by both viscosity and momentum forces, and therefore is a function of the shape, surface smoothness, size, and velocity of the body; the density and viscosity of the medium in which the body has motion; the speed of sound in the medium; and, lastly, the existing motion of turbulence of the particles of the medium before encountering the body.

The parasite or interference drag refers to drag of all parts other than the basic body, that is, external lights, bumpers, external antenna, wind wings, and the like. Their mutual interference, the interference between these items and the main body, contributes to the parasite drag of an automobile. The drag of these items is caused by the same

■ Fig. 3 – Power required to overcome air resistance plotted against speed at various drag coefficients

factors that produce drag on the basic body, plus the addition of interference. Also included under parasite drag is the drag of the internal flow of air through the body, such as radiator and ventilator duct losses. The interference, or change in flow around bodies in close proximity, from flow around the same bodies considered separately, may easily increase by 200% the drag of the basic body and the total parasite drag over the sum of these parts considered separately. In automobiles, there is an additional interference effect between vehicle and ground plane or surface of road.

In aerodynamics there are existing parameters that deal with these qualities. One of the most important of these is the effective Reynolds number which is the index of scale effect. This number is determined from the size and velocity of the body and the density, viscosity, and turbulence of the fluid. In order that model data may be applied to full-scale vehicles, it is necessary to duplicate the effective Reynolds number or make corrections from similar data. Another parameter, the Mach number, or the ratio of the speed of the body and the speed of sound in the fluid, is a measure of the compressibility effect. Fortunately, compressibility is a negligible factor at car speeds.

The value of the total drag of the vehicle could be thoroughly and practically checked in a wind tunnel. As the maximum Reynolds number of the full-scale vehicle is of the order of 18,000,000 (100 mph), the scale effect of the model should not be a serious factor. The model could easily be one-quarter size in a typical tunnel, with model Reynolds number of 9,000,000 represented by a tunnel speed of 200 mph. There is a number of tunnels throughout the land that have working chambers capable of receiving a complete full-scale automobile. Unlike wind tunnel tests of airplanes, which are supported well away from the walls and floor of the tunnel and are rotated in both pitch and yaw, the automobile model should be supported with its wheels almost touching the floor of the tunnel or ground plane and would be rotated only in yaw. Although the drag or force component parallel to the wind axis is of great interest, complete six-component data should

be taken in order to locate centers of forces and moments and to determine that the aerodynamic stability of the vehicle (as affected by cross wind, perhaps) is satisfactory.

Frequently, wool tuft tests can provide an excellent visual indication of the character of flow over and around the body, thus photographically determining the pattern. A representative model of a small passenger plane, about the size and power of a coupé, is shown in Fig. 2, mounted in a wind tunnel and undergoing flow investigation. The wool tufts are all over the body and wings, but those seen around the sides and top of the body and along the wings at the intersection are of particular interest in our discussion. Incidentally, the wings were at an angle of attack of plus 10 deg to the wind when this shot was taken. Therefore, the flow around the body and across the intersections was quite turbulent.

But what about some sort of approach to the problem of improved shapes and lines resulting from this science of airflow and drag? If we could only waive the precedent set by styling? Precedent may not always comply with the puritanical aerodynamic practice for low form drag, but does agree that the area and interior volume needs for current acceptance of seating capacity and comfort must be met.

Although in the preparation of this paper it seemed ill-advised to wind tunnel test a model or models of representative automobiles, nevertheless, the character of the definitive lines that can easily be developed in one's mind, were considered. Three eras of five-passenger, two-door sedans were visualized: (A) an abruptly lined, "boxy" type of the vintage of 1925 with a "supreme" 6-cyl engine of a generous 60 hp – a car worthy of its name. Then for (B) a 1945, five-passenger job with a nice broad shiny bridgework across the front and a silver streak in its tail. But the pay-off is (C). It has all the Hollywood sex appeal – lines like a Grecian urn; and it may have its preview after Congress gets the law of gravity repealed.

The drag of the car has been reduced to a mathematical coefficient called C_D, and the eras have been defined by Alpha, Beta, and Gamma and represented by a value of 1, 0.5, and 0.25, respectively. Fig. 3 shows the brake horsepower required to overcome air resistance versus speed at various drag coefficients. These are typical curves – power varying as the cube of the speed. They refer to cars, which, as Red Skelton would say, "If they have any resemblance to things living, they had better be dead." Obviously, for comparison, I have in the first instance assigned conservative to pessimistic values; and in the latter, very, very optimistic values of drag.

From this development, one comes to the sad but stern realization that unless one is talking about speeds of, say, 100 mph or better, the improvement between the best and the worst shape is rather discouraging, just like it is amongst the female of the species. But does that mean that nothing should or could be done by applying aircraft design practice to auto body design? It is the authors' private opinion that a host of interesting secondary design considerations of aerodynamic character would be found of value. They may be very mild and lacking in style appeal when compared with the set of gold teeth across the front of the cars or the fast backs that give the sleek, smooth airflow appearance at the possible sacrifice of head room. But, in any event, these are some of the details that an aircraft approach would probably reflect when covering the drag characteristics of a body.

The drag due to the recessed glass could be reduced by making windows and windshields flush. The degree of surface smoothness has become a recognized factor of importance as speeds go up. A detail that seemed worthy of attention is the drag of sharp surface irregularities, items which cause disturbances or turbulent flow of the air. Such things as abrupt change in section, reentrant angles where flow cannot be laminar but instead turbulent, resulting in a thick boundary layer, should be eliminated. The buffeting, which is drag as well as noise, is the result of improper intersections of the side posts with the windshield and the V of the windshield with the brow of the body where they meet.

You men in the motor car business may have heard that there are grave possibilities that some of us in the airplane business may go "belly up." Whether this comes to pass or not, it seemed of interest to draw your attention to the belly side of a plane and draw some comparisons with the ground vehicle. The streamlining or smoothing out of the flow along the bottom of an airplane body is important, so the aircraft man might logically conclude the same is the case with the car. Thus we might test the model with a smooth belly and then a typical rough one with axles, cases, shafts, pans, and so on, extending all along the bottom. The real problem is to be certain that with the smooth bottom the pitching moment of the vehicle is such that it would not tend to become light on the front wheels and to a lesser degree on the rear wheels and thus become unruly. As for the slope of the body as it sweeps back to the tail, it might well be that the severe slope or, as we heard it called, "the fast back" may not need to be so severe for low drag – for, after all, the flow over the crown of the top and the turbulence resulting therefrom may be such as not to increase drag by filling out curvature at back.

ENGINE INSTALLATIONS

Cooling problems connected with radial engine installations are thoroughly analyzed in a paper by Frederick V. H. Judd of United Aircraft Corp. entitled "A Systematic Approach to the Aerodynamic Design of Radial Engine Installations."

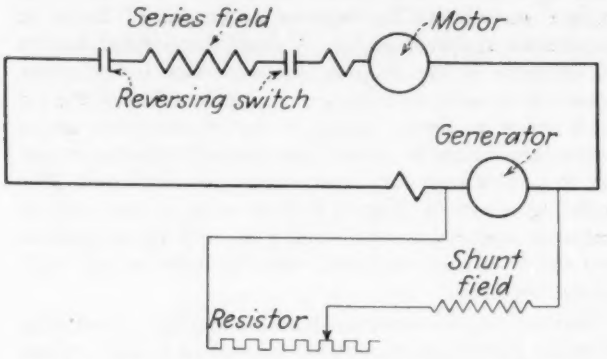
The major part of the report is devoted to the development of a general method for estimating the airflow through and drag of naturally aspirated radial aircooled engine installations. Charts are presented by means of which the optimum installation may be designed from an airflow and drag point of view. A detailed, step-by-step example of cooling analysis is also given.

The report brings up to date the published results of a series of wind tunnel tests of model aircooled engine nacelles that have been conducted by the Research Division of United Aircraft Corp. Results of new tests on the effects of operating propellers, variations in cowl flap length and circumferential extent, special types of cowl flaps, and tests of heat transfer from finned cylinders are included.

Due to its extreme length, this paper will not be published for several months, however, the full text, including illustrations, is now available from the Special Publications Department, Society of Automotive Engineers, Inc., 29 West 39 St., New York City 18, for 25¢ to members, 50¢ to nonmembers.

TRANSMISSIONS Gi

by P. M. HELDT



■ Fig. 1 – Simplified diagram of connections of electric drive

THE conventional or predominant drive for all types of motor vehicles during the past half-century has been the geared drive with which a number of different gear trains giving different transmission ratios are engaged successively. With such a drive the vehicle is accelerated from a standstill to its normal speed in a number of steps. Individual steps or acceleration periods are separated by an interval during which there is no acceleration, because a gear change can be effected only when the engine is disconnected from the drive by the friction clutch. During this period of disconnection, which, of course, is very brief, the vehicle actually decelerates.

These interruptions in the acceleration of the vehicle are of considerable importance in the case of city buses, as clearly brought out in a report of the Transportation & Maintenance Technical Committee.[1] In order that fairly fast schedules can be maintained in city operation, the average rate of acceleration (and deceleration) must be high, yet the maximum rate must not be so high as to cause discomfort to passengers. It is obvious that if there are no interruptions in the acceleration, the maximum rate can be less for a given mean value. The rate of increase of acceleration, or what is sometimes called the jerk, also is of importance, especially from the standpoint of standees, for although the force that tends to upset a standee is proportional to the acceleration, if the maximum acceleration is attained sufficiently gradually, he usually has time to brace himself against it.

The situation is rendered difficult by the fact that acceleration can increase and decrease exceedingly rapidly. We all know that it takes time to bring a large mass up to speed. The relation between the three factors velocity, time, and acceleration is given by the equation $v = ta$; that is, if the acceleration is constant, the velocity or speed increases in direct proportion to the elapsed time. For the acceleration we have the equation $a = F/M$; that is, the acceleration is equal to the quotient of the force by the mass on which it acts. Time is not a factor in this equation, and if it were possible to apply a definite force to a body instantly, the acceleration would instantly assume a definite value, and its rate of increase would be infinite. However, owing to the fact that all materials are elastic, it is impossible to apply a definite force instantly. Ordinarily, a force is applied to a body by bringing another body in contact with it. The force then exerted by the one body on the other causes elastic compression at the points or areas of contact, and during this period of elastic deformation the force applied increases from zero to the maximum. However, where relatively rigid materials are concerned and the stresses at the areas of contact remain within the elastic limit, the yield is extremely small. Hence, the rate of increase of pressure, and therefore of acceleration, is very great. In other words, the maximum acceleration, while not instantaneous, is reached in a very short time.

From the report of the Transportation & Maintenance Technical Committee[1] it is obvious that for city buses a type of drive or transmission that gives uninterrupted acceleration has definite advantages. It increases passenger comfort, reduces risks of personal injury to standees, and decreases wear and tear on all parts of the mechanism which are stressed by application of the power. Experience has shown that failure of parts of the driving mechanism, such as axle shafts and gears, are practically always fatigue failures; that is, they are brought about by a succession of localized stresses exceeding the elastic limit of the material at the point of failure. These stresses naturally occur when the torque transmitted reaches peak values, and if the torque peaks can be reduced, the lives of such parts will be increased.

Continuous acceleration can be obtained by mechanical, by electric, or by hydraulic means, or by a combination of mechanical with either electric or hydraulic devices. This paper will be devoted to a general discussion of these various types of transmission or drive.

In a vehicle propelled by a combustion engine the ratio between the rotary speeds of the engine crankshaft and the propeller shaft must be changed during the period of acceleration, and to achieve uninterrupted acceleration this change in ratio must be brought about without interrupting the flow of power. There are a number of mechanical transmission devices which permit a gradual change in the speed ratio between their input and output shafts, but it may be said right here that not one of these is at all promising as a bus transmission under modern conditions. The chief objection to most of them is that in order to be able to transmit successfully the power of an average bus engine, they would have to have such large overall dimensions that it would be impossible to accommodate them on a bus. These mechanical transmission devices may be briefly enumerated as follows:

1. Belt with conical pulleys (not stepped pulleys).
2. Belt with expanding and contracting pulleys.
3. Friction disc and wheel.

[This paper was presented at a meeting of the Philadelphia Section of the SAE, Philadelphia, Pa., Feb. 13, 1946.]

[1] See "Report on Importance of Steady Acceleration and Practical Value of It." Prepared by Transportation & Maintenance Technical Committee of the Transportation & Maintenance Engineering Activity, Society of Automotive Engineers, Inc., March, 1945.

Giving UNINTERRUPTED Acceleration

4. Grooved friction discs facing each other, with friction pulleys between them.

5. Variable-throw devices combined with roller ratchets.

All of these various devices were tried in the passenger-car and truck fields in the early years of the industry, when horsepowers were quite moderate, and they proved unsatisfactory even under those relatively easy conditions.

From the mechanical we pass to the electric drive, which in its modern form always gives uninterrupted acceleration. In the battery electric vehicles of an earlier period the speed quite often was controlled by connecting the batteries alternately in series and in parallel. This design entailed opening of the main circuit and interruption of the driving power. When the electric energy is generated on the vehicle, it is possible to use two motors and control the speed by connecting these either in series or in parallel, which also involves opening of the main circuit. In modern electric drives a single motor is used, as a rule. The motor is connected in circuit with the generator, and after the main switch or reversing switch has been closed to start the vehicle, the main circuit remains closed, and current from the generator flows through the motor throughout the period of acceleration. The only change made in the electric circuits while the vehicle is being accelerated is that resistances in the generator field circuit are cut out step by step. This, of course, can be done without interrupting the field circuit and the motor torque.

The effect of this method of control on vehicular acceleration is well shown by the acceleration diagram of the Presidents Conference Committee car in the report of the Transportation & Maintenance Technical Committee.[1] After the switch is closed, acceleration increases at a rapid, substantially constant rate, up to its maximum value. Then, as the generator field resistance is cut out step by step, the acceleration remains substantially constant, and after all of the field resistance has been cut out, the acceleration drops gradually in accordance with the speed-torque curve of the series motor, at first rather rapidly and gradually more slowly. That the acceleration decreases with increase in speed is quite natural, because more and more of the available power is required to maintain the speed, and less is available for acceleration.

A simplified diagram of connections of a modern electric drive is shown in Fig. 1. The drive comprises a shunt-wound generator and a series-wound motor connected in circuit with each other. In the diagram the main circuit, which is shown in heavy lines, will be seen to comprise the generator and motor armatures, the commutating field windings of both generator and motor, the series field winding of the motor, and the reversing switch. By means of the reversing switch the direction of current flow through the motor field can be reversed, which results in a reversal of the direction of rotation of the motor armature. When it is desired to start the vehicle, the reversing switch is closed, and it remains closed throughout the accelerating period.

The field circuit of the generator includes the shunt winding and a rheostat or resistor. Resistance in the field circuit can be cut out stepwise without opening the circuit.

In Fig. 2 is shown a typical torque-speed curve of an electric drive, corresponding to full-throttle operation. When the torque load on the motor is small, the latter will run at quite high speeds, but with increase in the torque load the speed decreases. If the engine ran at constant speed and delivered the same power throughout, and if the efficiency of the drive were constant throughout the speed range, the motor output would be constant, and the torque-speed curve then would have the form of a hyperbola. The hyperbola drawn, in Fig. 2, in dotted lines, corresponds to the mean between the minimum and maximum outputs shown by the speed-torque curve of the electric drive. In this case it happens to be equal to the output at maximum motor speed. The speed-torque characteristic of an electric drive is similar in form to the characteristic of a series motor supplied with current at constant voltage. The generator voltage of an electric drive is not constant, being lower at the low-speed, high-torque end of the range, but it varies only moderately.

Besides continuous or uninterrupted acceleration the electric drive has the advantage of ease of control. Operation of a bus with electric drive does not fatigue the driver as much as operation of a bus with an ordinary geared transmission, with which in city operation the clutch pedal must be depressed against a powerful spring at frequent intervals. The principal objections to the electric drive are its great weight and high cost. Weights range between 12 and 14 lb per engine hp for drives for between 100 and 200 hp. The efficiency of the electric

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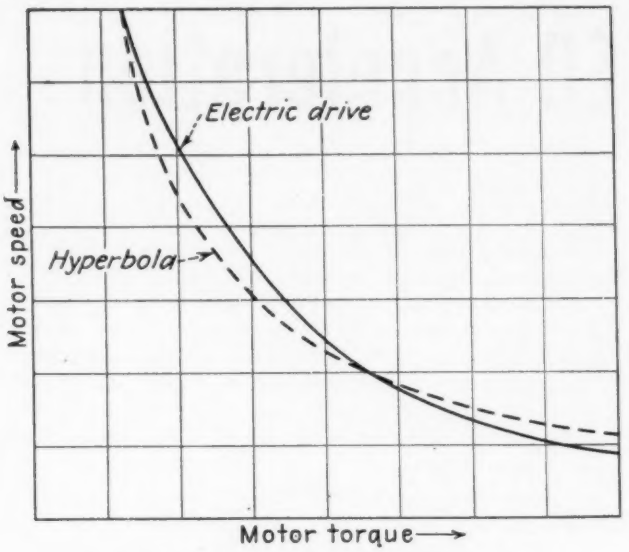
PARTICULARLY for city buses, as Mr. Heldt points out, there are definite advantages in using a type of drive or transmission that gives uninterrupted acceleration.

Continuous acceleration can be obtained by mechanical, electrical, or hydraulic means, or by a combination of mechanical with either electrical or hydraulic devices. Mr. Heldt devotes this paper to a general discussion of these various types of transmission or drive.

THE AUTHOR: P. M. HELDT (M '06) is the publisher and author of numerous widely used automotive text books, many of which have been translated into several languages. Before entering the editorial field in 1900, he was associated with the Munson Co., manufacturers of a combination gasoline-electric automobile, and Western Electric Co. He covered the Paris World's Fair of 1900 as editorial representative for *Electrical World* and *Horseless Age*. Returning to the United States he joined the editorial staff of the latter magazine (now Automotive Industries) where he served as engineering editor for more than 40 years.



■ Fig. 2 – Torque-speed curve of electric drive

drive at constant speed also is materially lower than that of a geared drive. However, under operating conditions where the vehicle is almost constantly being either accelerated or decelerated, the comparison is more favorable to the electric drive, because under these conditions in the vehicle with mechanical drive the clutch will slip, and the percentage of power loss in the clutch will be exactly equal to the percentage of slip. This loss, of course, must be added to that in the transmission proper in order to arrive at the overall efficiency of the drive.

A further advantage of the electric drive is that a reverse motion is obtained by very simple means – a reversing switch in the main circuit which reverses the direction of current flow through the motor field circuit.

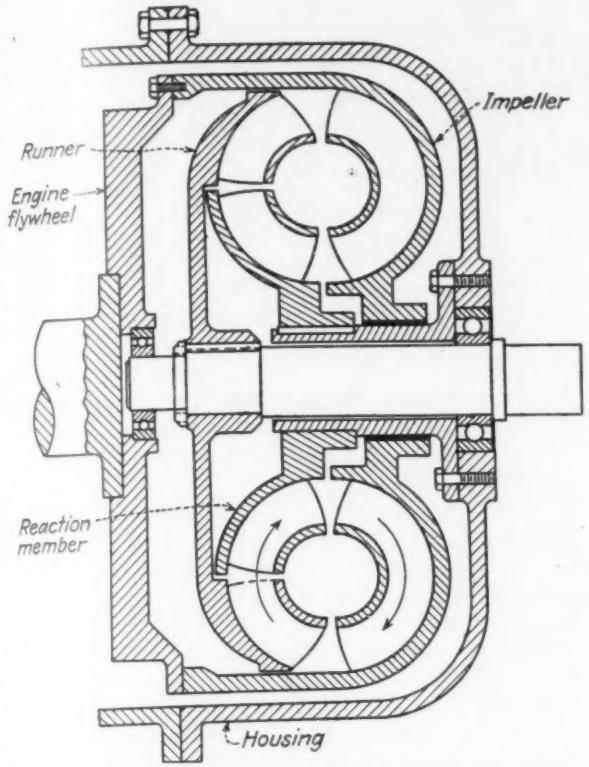
Uninterrupted acceleration can be obtained also by means of hydraulic transmissions of the positive-displacement type. These comprise a pump which places the hydraulic fluid under pressure and sets it in motion, and a motor which is operated by the moving fluid. Transmissions of this type are referred to as hydrostatic, though the term is not a very happy one, because with the device in operation the fluid is in motion and not static. The only justification for the use of the term seems to be that the moving parts of the motor are subjected to what is generally known as hydrostatic pressure. Variations in the torque ratio are obtained by changing the stroke of the pump plungers, by means of a swashplate of variable inclination, double eccentrics angularly adjustable, or some other suitable mechanism. When the pump displacement per revolution is decreased, the available engine power makes it possible to place the fluid under higher pressure, and fluid entering the hydraulic motor under higher pressure will cause it to develop greater torque. If it is assumed that in a transmission of this kind the pressure is uniform throughout, the ratio of output torque to input torque is equal to the ratio of motor displacement to pump displacement per shaft revolution.

Hydrostatic transmissions of this general type have been specially designed for use of motor vehicles, among them the Manly, descriptions of which have appeared in the technical press. But very little has been heard of them in recent years, apparently because they are poorly suited to high-speed operation. The various elements of these hydrostatic transmissions are comparatively simple, and hydraulic drives are quite satisfactory in industrial applications, where the velocities of fluid flow can be held low. In automotive transmissions it would be necessary (or at least quite desirable) to run the pump at engine speed, in order to keep down its bulk and weight, and plunger-type hydraulic pumps cannot be operated efficiently at such high speeds. The flow in the connecting passages between the pump and motor cylinders, as well as in the cylinders themselves, would be of a turbulent character, and in turbulent flow the losses vary as the cube of the velocity, whereas the power transmitted varies only as the first power of the velocity, assuming the pressure to be constant. This explains why the efficiency of such transmissions drops rapidly with an increase in the operating speed.

Another type of hydraulic transmission permitting uninterrupted acceleration is the hydrokinetic torque converter. It is similar in principle to the hydraulic coupling, which has become familiar in the automobile world in this country under the popular name "fluid drive" and in England as the "fluid flywheel." The coupling consists of a housing enclosing two vaned rotors, one, known as the impeller, being secured to the driving shaft, and the second, known as the runner, secured to the driven shaft. With only rotors in a stationary, smooth-walled housing the torque on the driven shaft is always the same as the torque on the driving shaft. In order to get an increase in torque on the driven shaft it is necessary to have an extra set of stationary vanes to take torque reaction. The reaction to the torque impressed on the impeller is taken by the engine or other source of power, and if there is to be additional torque on the output shaft, means must be provided for taking the reaction due to it. In its simplest form, therefore, the hydrokinetic torque converter consists of a housing enclosing an impeller which sets the fluid in motion, a runner set in rotation by the moving fluid, and a set of stationary vanes which take the reaction due to the excess of torque on the runner over that on the impeller. Such a converter is shown diagrammatically in Fig. 3.

Although basically the only difference between the hydraulic coupling and the converter is that in addition to the impeller and runner of the coupling the converter has a set of stationary reaction vanes, there are many important differences of detail design between the two devices. In the coupling the vanes of both impeller and runner are radial. This is permissible because in normal operation both members turn at nearly the same speed, and the fluid therefore can pass from the impeller to the runner without severe shock. In a converter, on the other hand, there is normally a large velocity differential between adjacent sets of vanes. For instance, under conditions of heavy traction, or during acceleration, the runner may be turning at only one-third the velocity of the impeller, and with radial vanes in both members the fluid under these conditions would be subjected to severe shock when entering the runner from the impeller, which would result in high turbulence and low efficiency. Steps taken in the design of the hydraulic torque converter to reduce the internal losses may be explained by reference to Fig. 4, which is a cutaway view of the Lysholm-Smith torque converter as built by the Twin Disc Clutch Co.

The impeller vanes *P* are made to slope backward from



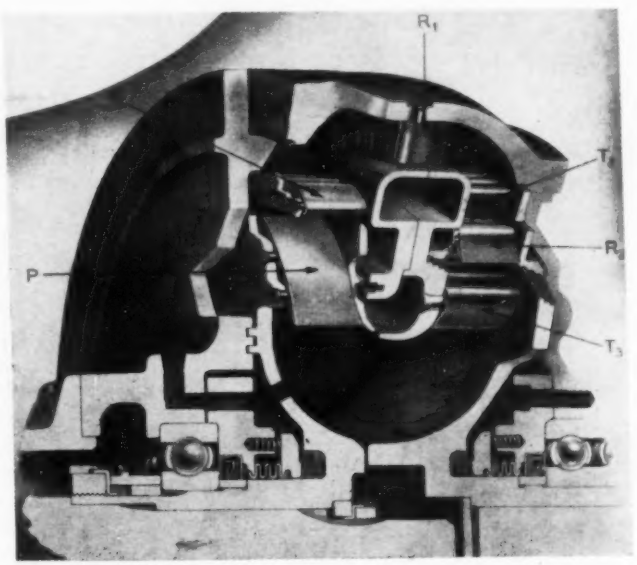
■ Fig. 3 – Diagram of single-stage hydrokinetic torque converter

the axis of rotation outward, in accordance with practice in the design of centrifugal pumps. Then, instead of the runner having only a single set of vanes, it is provided with two or more sets (three in this case). In Fig. 4 the three sets of runner vanes are denoted by T_1, T_2, and T_3. Rows of reaction vanes R_1 and R_2 are interposed between successive rows of runner vanes. Both the runner vanes and the reaction vanes are given an airfoil section, and all vanes are set at such angles as to ensure the most favorable angle of incidence of the fluid on the vanes. Reaction vanes are riveted directly to the housing, while the runner vanes are secured to an inner core member and to the runner disc.

A typical torque-speed curve of a hydrokinetic torque converter is shown in Fig. 5. Speed and torque of the runner are given in percentages of impeller speed and torque, respectively. The hyperbola shown in dashed lines corresponds to the maximum output of the converter, which is attained when the runner turns at about 40% of impeller speed. The curve shows that with the runner at rest the torque on it is about 5.3 times the impeller torque, and that the output torque drops to zero when the runner turns at about 90% impeller speed. Both the maximum relative torque and the maximum relative speed depend on details of design and certain qualities of the fluid used, particularly its viscosity. It is doubtful whether as high a torque ratio as 5.3 can be obtained with a converter with a single-stage runner, but it has been obtained with converters having multistage runners. It will be noticed that the speed-torque curve of the converter drops away sharply from the ideal hyperbola at both ends of the range. This is due to the fact that the efficiency decreases greatly toward the ends of the range.

A typical efficiency curve for hydrokinetic torque converters is shown in Fig. 6. The efficiency of transmission reaches a maximum value of about 84% at a speed-reduction ratio of 2.5 (runner speed = 40% impeller speed). When starting from a standstill (zero runner speed) the efficiency naturally is zero, because when the speed is zero the output is zero. Typical efficiency curves for a hydraulic coupling and a straight electric drive also are shown in Fig. 6. In the hydraulic coupling the losses are equal to the percentage of slip, and the efficiency in per cent therefore is equal to 100 minus the percentage of slip. Under normal road load the slip amounts to only a few per cent, and the efficiency then usually lies between 97 and 98%. It will be noticed that whereas both the hydraulic coupling and the electric drive are at practically their maximum efficiency when the speed ratio is close to 1:1 (direct drive), with the hydraulic torque converter the efficiency falls off rapidly when the speed ratio decreases below 2:1 (50% on the axis of abscissas). While the maximum efficiency of the hydraulic converter is somewhat higher than that of the electric drive, when the ratio of output speed to input speed exceeds 0.60, the efficiency of the hydraulic converter drops below that of the electric drive, and at a speed ratio of 0.75 it is only about one-half as large. As it is usually desired to have the propeller shaft turn at about engine speed in normal operation, it is evident that the hydraulic torque converter cannot be used throughout the operating range, but it is quite efficient for acceleration over the first 60% of the speed range. When a speed of about 60% of that corresponding to direct drive has been reached, the converter action must be cut out and a type of drive substituted which is more efficient in this higher speed range. The driven shaft may then be either connected to the driving shaft by a friction clutch, to give a positive direct drive, or the converter may be transformed into a hydraulic coupling.

The first principle is embodied in the Lysholm-Smith hydromechanical transmission which originated in Sweden and is being manufactured in this country under license by the Spicer Mfg. Corp. and the Twin Disc Clutch Co. Transmissions of this type have been used especially on Yellow Coaches. The second principle is embodied in the



■ Fig. 4 – Cutaway view of Twin-Disc torque converter

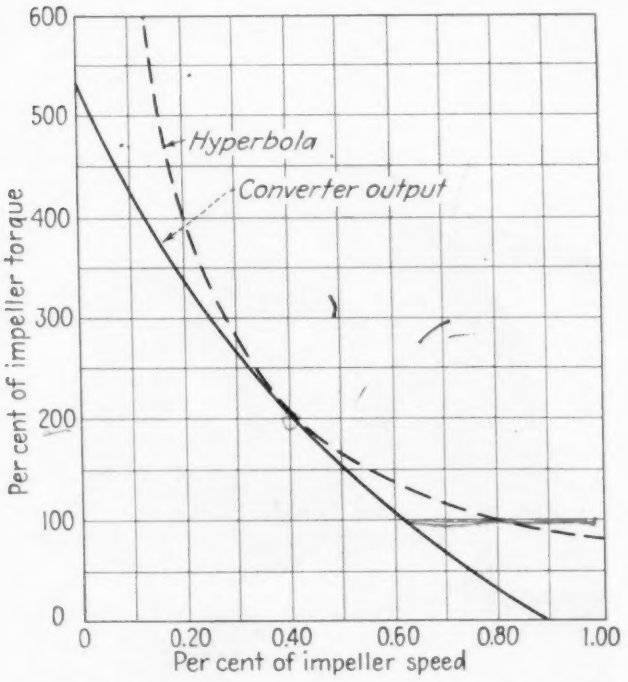
Trilok transmission which originated in Germany, where it was first announced in 1934.

Assuming the efficiencies of the different units to vary as shown in Fig. 6, if the change-over from hydraulic converter action is made when the runner turns at 62% impeller speed, where the converter efficiency is about 62%, the efficiency of transmission will thereafter increase directly with the speed ratio, up to 97 or 98% in the case of the hydraulic coupling, and to 100% in the case of the friction clutch. The transmission with friction clutch, therefore, has the advantage of a slightly higher efficiency at normal speeds, while that with the hydraulic coupling has the advantage of greater flexibility or smoothness within this speed range.

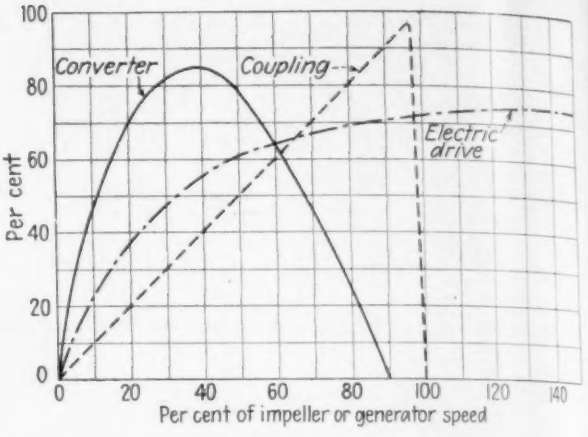
A hydraulic torque converter does not provide a reverse speed, and a special reversing gear must be combined with it. Also, when the direct drive is used the hydraulic torque converter must be cut out, and vice versa. For this purpose two friction clutches are now provided, combined in a single unit and operated by a single lever. With the lever in one extreme position the flywheel of the engine is connected by friction clutch to a shaft extending through the hollow impeller and runner shafts and coupled to the propeller shaft.

In the Lysholm-Smith hydromechanical transmission as used on Yellow Coaches (Fig. 7), the runner of the torque converter has three rows of vanes, and these are separated by two rows of stationary guide vanes. With multistage runners the angle of incidence of the fluid on the vanes can be made smaller, hence the flow is smoother and losses due to shock are smaller.

When a hydraulic coupling is used to transmit the power within the higher speed range, it is not necessary to provide a separate device, as the converter can be made to act as a coupling under these operating conditions. This requires the provision of two roller clutches or one-way



■ Fig. 5 – Torque-speed curve of hydrokinetic torque converter



■ Fig. 6 – Efficiency curves

clutches. Fig. 8 is a sectional view of the Trilok drive which was brought out in Germany in 1934 by the firm of Klein, Schanzlin & Becker Co. S_1 is the driving shaft, I, the impeller, R, the runner, and G, the reaction member or series of guide vanes. The guide vanes are mounted on a tubular shaft surrounding the output shaft S_2. At its outer end the hollow shaft carries the inner race of the roller clutch C_2, the outer race of which is mounted in the housing in such a way that it cannot rotate therein. While the roller clutch allows the reaction member to rotate with the impeller and runner, it prevents it from rotating in the opposite direction. Therefore, while the unit acts as a converter the reaction vanes are stationary. A second roller clutch C_1 prevents the reaction member from rotating faster than the runner. When the runner torque decreases to the value of the impeller torque there is no longer a reaction on the guide vanes. Any further slight increase in runner speed will result in a reaction on the guide vanes in the opposite direction, and the reaction member will then rotate with the impeller and runner. Under these conditions, since the torque on the runner is the same as that on the impeller, the device functions as a hydraulic coupling.

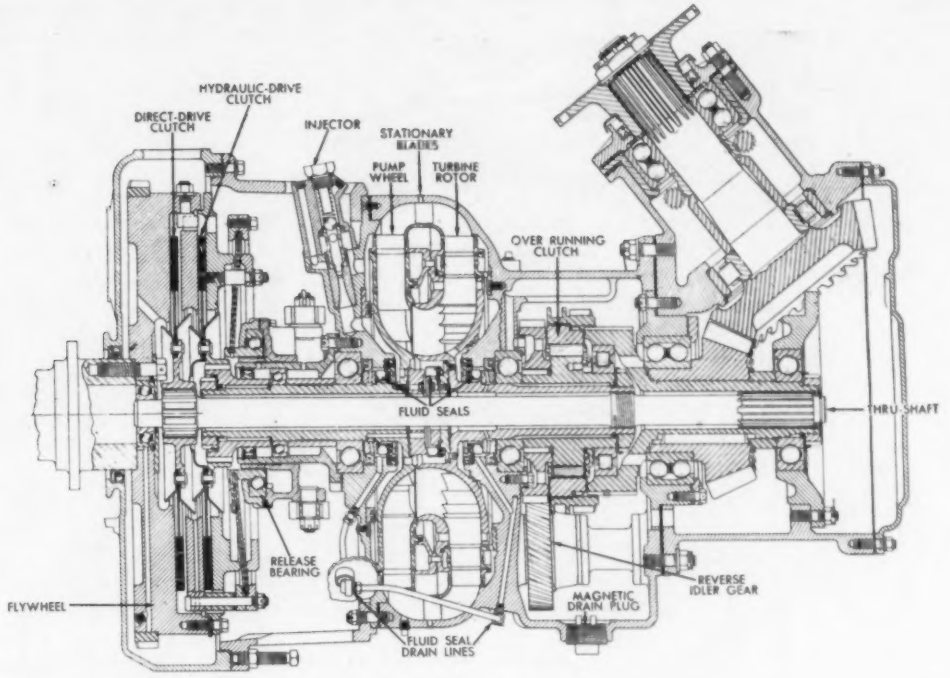
It can be readily shown that the change from converter to coupling action occurs automatically when the torque-multiplication ratio due to the converter becomes equal to the reciprocal of the converter efficiency. The torque-multiplication ratio, of course, is at its maximum when the vehicle is being started, and it drops continually as speed is being gained. Let:

T = Impeller torque
N = Impeller speed
r = Speed ratio
t = Runner torque
η = Efficiency of converter

Then the power input is $2\pi T N$ ft-lb per min and the output:

$$2\pi N T \eta = 2\pi \frac{N}{r} t$$

Now, when the output torque is equal to the input torque, $t = T$, hence η must be equal to $1/r$. In Fig. 6 the efficiency curves of a hydraulic torque converter and a hydraulic coupling are superposed on a base of speed-ratio reciprocals. That is, ordinarily we think of the speed



■ Fig. 7 – Yellow-Coach heavy-duty hydraulic transmission

ratio as the ratio of impeller shaft speed to runner-shaft speed, while the figures plotted along the base line in the diagram are ratios of runner-shaft speed to impeller-shaft speed. It will be seen that when the reciprocal of the speed ratio is 0.625, which corresponds to a speed ratio of 1.6, the runner torque becomes equal to the impeller torque and the device therefore begins to act as a hydraulic coupling. Thereafter, the reaction unit turns with the impeller and runner.

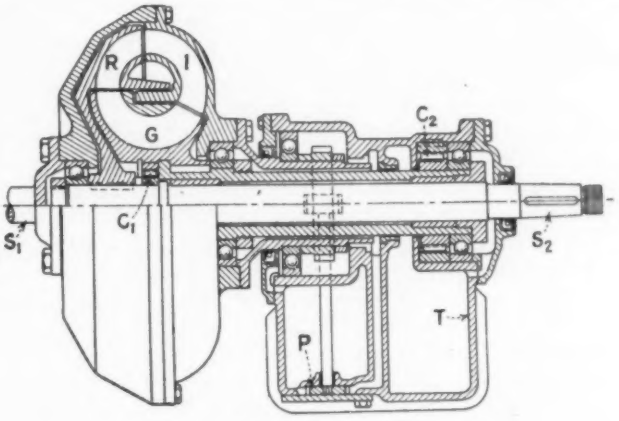
The view has been expressed that when such a three-rotor unit operates as a hydraulic coupling, its efficiency must be lower than that of a two-rotor coupling. While there may be something to this contention, it is not likely that the difference in efficiencies is important. With the three-rotor as with the two-rotor coupling the output torque is always equal to the input torque, and if the three-rotor coupling is less efficient it would mean that for a given input torque the percentage slip is somewhat greater.

As already pointed out, the performance of a hydraulic converter can be improved by providing it with a multi-stage runner, and a number of rows or rings of reaction blades. A combined hydrokinetic converter and coupling having a two-stage runner has been developed by J. Brookham & Co., Ltd., of East Bromwich, England, and an axial section through this unit is shown in Fig. 9. I is the impeller, T_1 and T_2 are the two stages of the runner, and R_1 and R_2 the two rows of reaction vanes. Here, too, the reaction member is provided with roller clutches or free-wheeling units which enable them to rotate with the runner when operating as a coupling, but not faster than the runner.

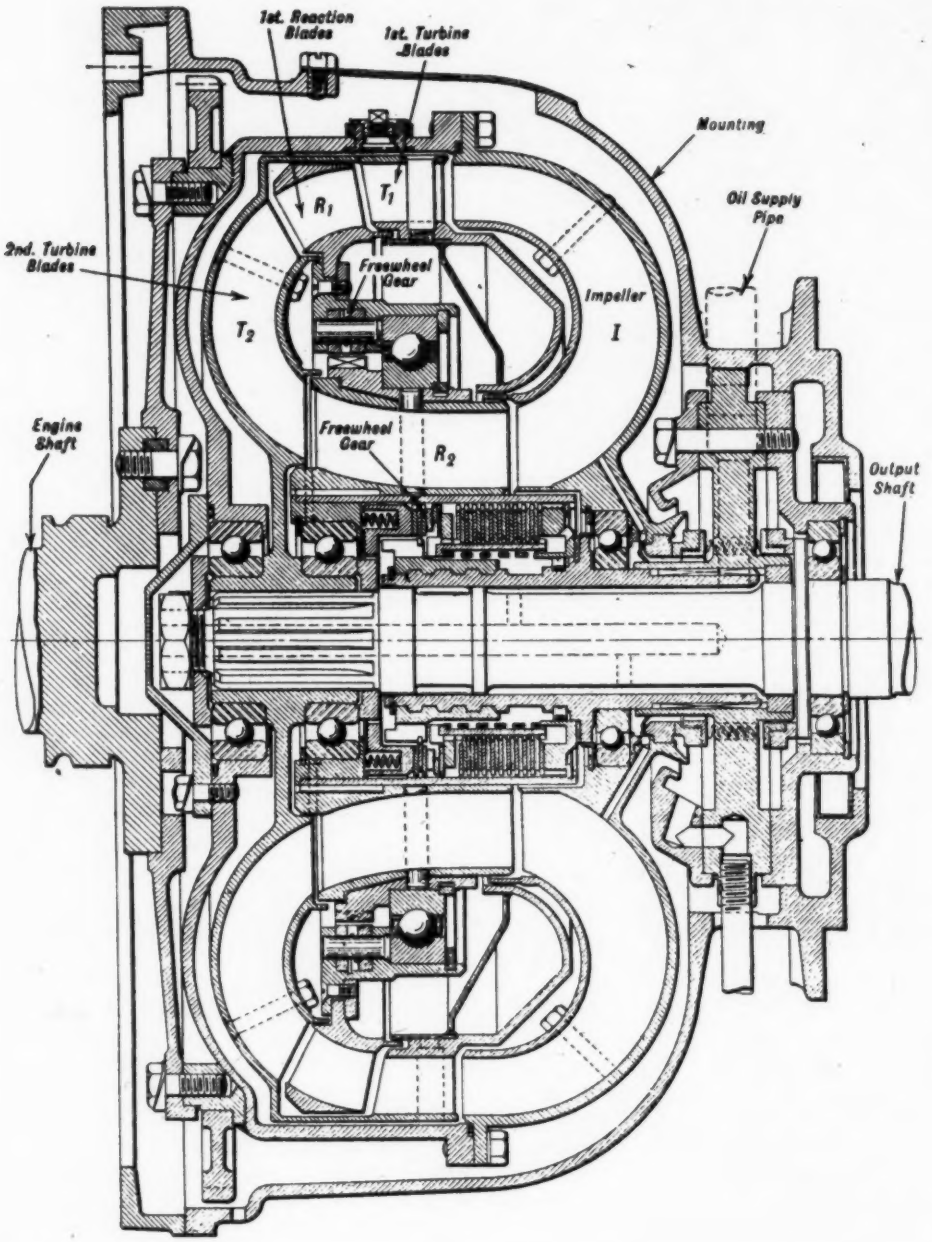
■ Differential Transmissions

Another type of transmission which permits uninterrupted acceleration is the differential type, of which quite a number of different forms have been worked out. A characteristic of this type is that the engine power is divided into two parts. One part is transmitted directly to the propeller shaft, while the other is converted into either electric or hydraulic power and reconverted to mechanical power, in which form it is combined with the part transmitted directly. The proportion of power transmitted directly varies automatically with the vehicle speed, and the torque-conversion ratio in what may be called the secondary power circuit, comprising the electric or hydraulic units, also varies automatically.

In a transmission of this type it is necessary first to divide the engine power between the two circuits or paths. This can be done mechanically by means of a differential or planetary gear; electrically by means of a direct-current generator, both the field frame and the armature of which can revolve, or hydraulically by means of a pump, both the cylinder block and the crankshaft or rotor shaft of which can revolve. Such a power divider naturally must have one power inlet and two outlets. In the case of a



■ Fig. 8 – Trilok hydraulic transmission



■ Fig. 9 – Brookham combination torque converter and coupling

differential or planetary gear there are three shafts, while in the case of the electric power divider one of the outlets is an electric cable carrying the current generated, and in the case of an hydraulic divider, a pipe or passage through which the fluid displaced by the pump flows.

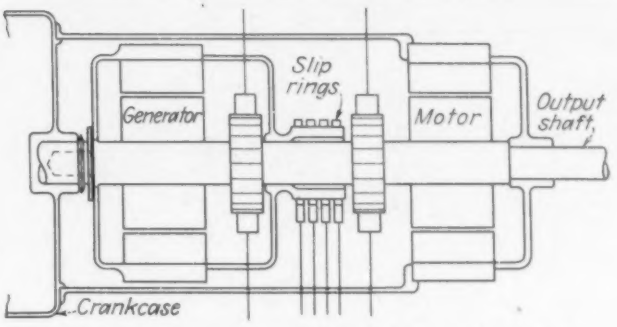
One of the earliest differential transmissions to come into practical use was the Entz, which was fitted to the Owen magnetic car. There the crankshaft power was divided into two parts in a very simple manner. Referring to Fig. 10, the field frame of the generator is flange-bolted to the engine crankshaft, and the armature of the generator is secured to the output shaft. In operation, the armature circuit of the generator being closed through the electric motor, current is generated in the generator armature windings and the revolving magnetic field then produces a drag on the armature, with the result that torque is transmitted directly from the engine crankshaft to the output shaft. The torque thus transmitted is equal to the engine torque, but since the propeller shaft or output shaft rotates at a slower speed than the crankshaft, not all of the engine power is thus transmitted.

The remainder of the engine power, disregarding losses, is converted into electric power. Electric current flows from the generator to the motor, whose armature is mounted on the output shaft, and the torque developed by the motor is added to that impressed on the output shaft by the magnetic drag of the generator. Under conditions of light traction, when the output-shaft torque required is within the limits of the engine torque, the output shaft will be turning at nearly engine speed. There is then little relative speed between the field and armature of the generator, and the electromotive force produced by

the generator and the current delivered by it are small. Since the electric motor then delivers only little current, it develops only little torque. On the contrary, when the car is being accelerated and is still running at low speed, the relative speed between generator field and armature is great, hence the current flow from generator to motor and the torque developed by the motor also are high. As the maximum engine torque is transmitted directly to the output shaft by the magnetic drag of the generator field on its armature, and the motor develops additional torque, the equipment serves as a torque converter, and the ratio of torque conversion varies automatically with the speed of the vehicle.

It is possible to design hydraulic equipment to operate on the same general principle, and at least two such transmissions have been developed, one (the Hall) in England a half-century ago, the other (the Schneider) in Switzerland more recently. Plunger-type pumps with radial cylinders were used in both of these converters. In the Hall the pump plungers were operated by an eccentric shaft and the cylinder block was connected to the output shaft. With the output shaft at rest, the plungers reciprocated in the pump cylinders and displaced fluid which was led into the cylinders of a hydraulic motor of the reciprocating type, whose cylinder block formed a rigid unit with that of the pump. The plungers of the hydraulic motor connected to a nonrotating double eccentric, and fluid forced into the motor cylinders by the pump produced a torque on the cylinder block of the motor, in the same direction as that on the cylinder block of the pump. The stroke of the motor plungers could be varied by means of the double eccentric and the ratio of torque conversion thus varied at will.

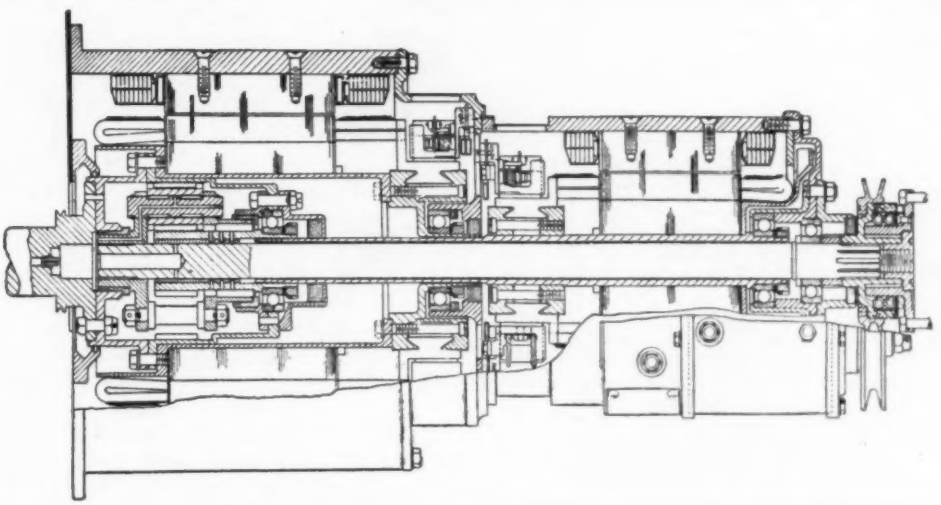
The principle of the Schneider was substantially the same, but the details were worked out entirely differently. The pump cylinders were reciprocated by eccentrics on the driving shaft and the pistons or plungers remained at a constant distance from the driving shaft axis, being attached to the cylindrical housing. The motor was of similar design, and the housings of both pump and motor were geared to the output shaft. I am not showing any illustrations of these transmissions and do not describe them in detail because I feel that since the pumps and motors are of the reciprocating type they do not lend themselves to high-speed operation.



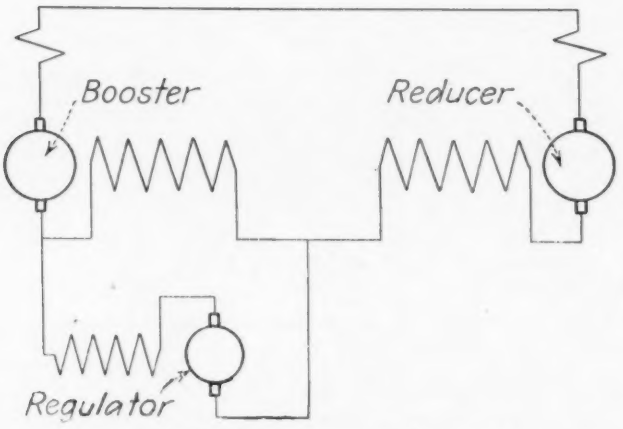
■ Fig. 10 – Diagram of Entz differential electric transmission

An electric differential transmission in which the engine power is divided by a differential gear of the planetary type is the electrogear, of which an axial section is shown in Fig. 11. It comprises as its principal components two series-wound dynamos with stationary field frames. Armature and field windings of both machines are connected in a single circuit. One of the electric machines, known as the booster, has its armature mounted on the engine crankshaft and serves as a flywheel for the engine, while the armature of the other electric machine, the reducer, is connected to the booster armature and the crankshaft through a planetary gear. In the particular design shown this planetary assembly consists of a ring gear secured to the booster armature, a sun gear secured to the reducer armature, and a number of planet pinions mounted on a spider or planet carrier which is secured to the output shaft. In addition to the two electric machines shown in Fig. 11, the equipment includes a third, of smaller size, known as the regulator dynamo. The latter, which is also series wound, is driven from the output shaft or propeller shaft, so that its speed is always proportional to the speed of the vehicle. As shown in Fig. 12, the regulator dynamo is connected across the field winding of the booster dynamo.

Now let us assume that the vehicle is stationary and the engine idling. The armature of the booster dynamo then revolves with the crankshaft at idling speed. Since the planet carrier secured to the propeller shaft is at rest, the planet pinions revolve around their studs and transmit



■ Fig. 11 – Axial section of electrogear

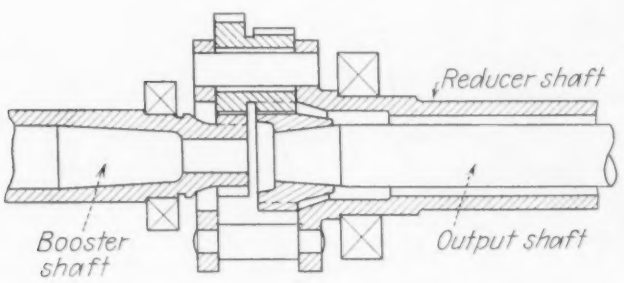


■ Fig. 12 – Circuit diagram of electrogear

motion to the sun gear, which then turns in the opposite direction to the crankshaft, at a considerably higher speed, in the ratio of the number of teeth in the ring gear to that in the sun gear. Since the reducer armature rotates much faster than the booster armature, the reducer generates a higher voltage than the booster, with the result that current generated in the reducer flows through the booster in opposition to the electromotive force generated by the latter and causes it to act as an electric motor. At idling engine speeds, however, the torque on the output shaft is not sufficient to move the vehicle.

When the accelerator pedal is depressed and the engine speeded up, the speeds of both armatures are increased, a heavier current is sent by the reducer through the booster, and the larger torque thus produced by the booster is added to the engine torque, so that at the start a rather high torque multiplication is effected, which results in rapid acceleration of the vehicle.

From the diagram of connections, Fig. 12, it will be seen that the regulator forms a shunt across the booster field, and while the vehicle is at rest or proceeding very slowly, part of the current from the reducer flows through the regulator, bypassing the booster field. The regulator, however, is driven in such a direction that it generates an electromotive force which is opposed to current flow through the regulator field, with the result that as the speed of the vehicle increases, the regulator field is first weakened and later reversed. Any increase in vehicle speed results in a weakening of the booster field, because, with a fixed engine speed, the higher the vehicle speed the lower will be the reducer speed and the smaller the current which it sends through the booster field. But if this effect were depended on alone for speed regulation, the maximum speed of the vehicle would be too low.



■ Fig. 13 – All-spur type of differential used with bus type of electrogear

At a certain vehicle speed, known as the equilibrium speed, the emf of the regulator is exactly equal to that impressed on the booster field by the reducer, so that there is then no current flow through the booster. At that speed, while some of the engine power is converted into electric power to maintain field magnetization, none of the electric power is reconverted into mechanical power.

With further increase in vehicle speed the regulator electromotive force increases beyond that of the reducer, and current flow through the booster field is then reversed. Since the booster continues to rotate in the same direction, it now acts as a generator, sending current through the reducer. Up to this point the reducer armature has been rotating in the direction opposite to that of the booster armature, but now it reverses in direction and turns in the same direction. With both the booster armature and the reducer armature turning in one direction, the planet carrier turns in the same direction. When the speed of the reducer armature becomes equal to that of the booster armature, all three members of the planetary gear have the same speed of rotation, and under these conditions a direct drive is obtained, the propeller shaft rotating at the same speed as the engine crankshaft.

If the traction resistance is sufficiently light, the reducer armature will attain a speed higher than engine speed, and the propeller shaft also will rotate at a higher speed than the engine crankshaft, so that an overdrive is produced automatically. Under these conditions the regulator develops a high electromotive force, which produces a strong booster field. This in turn results in a high booster voltage and speeds up the reducer armature, which then turns faster than the engine crankshaft and ring gear, and causes the planet-carrier to turn faster.

With the electrogear a reverse speed is obtained in much the same way as with a straight electric drive. Both the booster and reducer series fields are reversed by means of reversing switches, and the regulator is disconnected. The booster then acts as a self-excited generator and sends current through the reducer. The reducer then acts as a motor rotating in the direction opposite to the direction of crankshaft rotation.

In an electrogear specially designed for bus installation the ring-gear type of differential was replaced by an all-spur type, as illustrated in Fig. 13. With the ring-gear type the maximum practical reduction ratio obtainable between ring gear and planet carrier is about 1.8. This ratio determines the maximum torque multiplication obtainable, which was found to be insufficient for bus work. With the ring-gear type of differential the latter was placed inside the booster armature, which necessitated the use of a hollow, drum-like armature shaft. The spur type of differential is located between the booster and reducer armatures, which permits the use of a booster armature shaft of much smaller diameter. The shaft is still made tubular, because its outside diameter must be larger than the outside diameter of the differential pinion carried by it, so that a ball bearing mounted against a shoulder on the shaft can be stripped over the gear.

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Effect of Fuel Properties on Diesel-Engine Performance

by F. G. SHOEMAKER and H. M. GADEBUSCH

Detroit Diesel-Engine Division
General Motors Corp.

INTENSE commercial development of new refining processes during the war years has caused a complete reorientation of the liquid fuel situation.

Large-scale increase of catalytic cracking and hydrogenation facilities have considerably widened the spread of available products, and even synthetic manufacture of liquid fuels from either natural gas or coal has now become an entirely practical reality.

For the first time in the history of internal-combustion engines, the fuel problem has advanced into the realm of almost unlimited possibilities, thus stimulating wide interest in the direction in which further engine development may progress if fuels of almost any desired characteristics are made commercially available.

This changed situation is of particular significance in the diesel-engine field. Ever since their inception, compression-ignition engines have been considered by the oil industry as a very convenient type of prime mover thriving on the utilization of liquid fuels for which no better use could be found.

Partial revision of this opinion became necessary soon after the arrival of high-speed diesel engines in automotive installations. Operating experience with this equipment proved some types of fuel oils to be superior to others for satisfactory longtime engine performance, and a considerable amount of effort has since been directed toward the uncovering of those fuel properties which affect engine operation.

Fuel Oil Properties

Gas oils, such as generally supplied as diesel-engine fuels, are normally identified by a number of physical and chemical inspection data intended to ensure uniformity during the refining process.

Among these properties are found gravity, viscosity, flash, fire and pour points, distillation characteristics, carbon residue, sulfur content, water, sediment, ash, neutralization number, and color.

According to their possible significance for engine use, these specifications may be grouped as shown in Table 1.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 10, 1946.]

Viscosity and pour point ensure fluidity of the fuel at all temperatures, sediment and ash show the percentage of inorganic contamination, and water as well as neutralization number are a measure of unfilterable contaminating materials.

The heat content of a fuel oil is determined by the carbon and hydrogen content and is closely related to its gravity.

Combustion residue may be judged from the distillation recovery and color of a fuel oil, both of which are indicative of its content of undistilled residual material. A direct measurement of the combustion residue is given by the Conradson carbon test and the sulfur content.

The distillation temperature range is representative of the fuel's vaporization characteristics, and the flash and fire points indicate its inflammability.

Ignition Quality

From the conditions under which ignition occurs in a compression-ignition engine, it becomes obvious that neither the flash nor the fire point of the fuel can be of any significance, since both are determined by ignition from an outside source.

In search of a proper expression for the self-ignition quality of diesel fuels, numerous methods have, therefore, been employed in the past.

Direct measurement of the spontaneous ignition temperature by bringing fuel drops in contact with a hot plate was found to be unsatisfactory, since the droplet size as well as the material of the plate affect the results in wide limits.

Since practical experience has shown fuels of paraffinic composition to produce smoothest diesel-engine combustion, the paraffinicity of the fuel has also been used as criterion of its ignition quality.

Viscosity/gravity constant, aniline number, and a combination of aniline number and gravity known as "diesel

THE studies discussed here suggest that the power developed in a diesel is directly proportional to the heating value of the fuel and is independent of its cetane rating, volatility, chemical composition, and molecular structure. Both cetane and volatility, however, the authors say, have marked effects upon cold starting and light-load operation, when the ignition capacity of the cylinder is reduced.

THE AUTHORS: F. G. SHOEMAKER (M '20) and his co-author are writing their bylines in the *SAE Journal* for the second time in three months. Mr. Shoemaker, who has been active in the development of the 2-cycle diesel engine, is chief engineer of General Motors' Detroit Diesel Engine Division. H. M. GADEBUSCH, also a General Motors engineer, is a member of the engineering staff of the Detroit Diesel Engine Division.

Table 1 – Significance of Fuel Oil Inspection Data

Fluidity	Contamination	Heat Content	Burning Characteristics	Combustion Residue
Viscosity	Water and sediment	Gravity	Distillation range	Distillation recovery
Pour point	Ash		Flash point	Color
	Neutralization number		Fire point	Conradson carbon
				Sulfur

index" have served as indirect measures of paraffinicity.

Since satisfactory correlation of these expressions with actual engine performance could not always be obtained, an engine test method comparable to the well-known octane-number determination for gasolines was developed.

By this method the duration of the ignition delay period of a fuel oil is compared with the delay period of blends of two reference fuels, cetane and alphamethyl naphthalene.

The ignition quality of a fuel is rated in cetane numbers according to the percentage of cetane contained in the blend producing the same ignition delay as the fuel under test.

To permit an idea of the relative standing of the three most frequently used terms, a comparison of diesel index, cetane number, and spontaneous ignition temperature for a number of different fuels is given in Fig. 1.

Although a reasonably good correlation of cetane number and spontaneous ignition temperature can be obtained, it is obvious that the diesel index of a fuel oil is of no more importance than to ensure uniformity of different shipments from the same source.

While vaporization and self-ignition characteristics of diesel fuels can thus be rated in some manner, little is so far known about the factors governing the combustion process.

■ Diesel-Engine Combustion

Since both power output and fuel consumption of an engine must be closely related to its combustion process, a short analysis of the combustion in a compression-ignition engine may be in order. Cylinder pressure observations have shown that transformation of the fuel's heat content to mechanical power takes place in three distinct phases, as shown in Fig. 2.

For a certain time element after the beginning of the fuel injection the cylinder pressure continues along its normal compression cycle. During this period of ignition delay the combustible mixture is prepared by vaporization of the fuel droplets as they penetrate into the air charge.

Following the spontaneous ignition of the fuel vapor, a period of rapid pressure rise can be observed, which terminates in a peak appearing shortly after the piston has passed the top dead center.

Finally, the remainder of the injected fuel continues to burn at a slower speed for a considerable period of time until polytropic expansion of the charge begins.

From knowledge of these three phases as revealed by indicator diagrams, it can be seen that ease of vaporization and ignition appear to influence the delay period. Further progress of the combustion must be governed by other factors, particularly by the chemical composition of the fuel which thus enters into the discussion as a third major factor.

[1] See SAE Journal (Transactions), Vol. 30, May, 1935, pp. 191-197 + (disc.) 197-199: "Cylinder Events Studied in Logarithmic Diagram," by A. T. Gregory.

Investigations of the effects of fuel compositions are greatly complicated for two reasons. First, because of the fact that natural fuels are composed of numerous molecularly different components; and second, on account of the intrinsic relation of their physical properties to their chemical composition.

Thus, it is possible to ascribe a given engine operating characteristics to any one of several related fuel properties, only one of which can have actually influenced the results.

■ Full-Load Performance

In view of the changed fuel supply situation and the importance of better understanding of the effect of the three major fuel properties on the combustion process, a series of tests were conducted on a multicylinder, direct-injection, two-cycle diesel engine of General Motors manufacture.

To measure the prevailing cylinder pressures, one of the cylinders was equipped with a balanced diaphragm Research-type indicator.

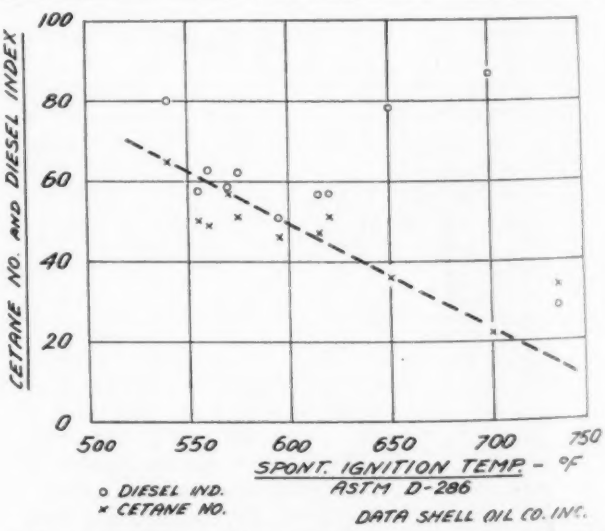
Beginning as well as ending of combustion were obtained from a logarithmic plot of absolute pressure versus cylinder volume (Fig. 3). The abrupt change in slope of the compression and expansion lines indicates the points at which combustion begins and ends.[1]

Brake horsepower and fuel consumption were measured in usual dynamometer practice and indicated horsepower values were obtained from the indicator diagrams as well as from the motoring friction.

■ Test Fuels

Twelve test fuels were procured containing straight-run as well as thermally and catalytically cracked materials and ranging in ignition quality from 22 to 65 cetane number and in volatility from 432 F to 606 F at the mid-boiling point.

Their composition included some predominantly paraffinic fuels as well as various mixtures of paraffinic, olefinic, aromatic, naphthenic, and isoparaffinic constituents, such as obtained from natural stocks by different refining processes. In addition, pure cetane was employed for some of the tests.



■ Fig. 1 – Variation of cetane number and diesel index with spontaneous ignition temperature

Table 2 – **Fuels Used to Determine Effect of Cetane Number**

Cetane Number	Volatility (Final Boiling Point), F	General Composition	Olefins (Bromine No.)	Aromatics (Specific Dispersion)
57	649	P	1.4	109
51	638	P,N,A	2.1	122
46	610	P,A,IP	2.1	137
36	642	P,O,IP	18.1	107
22	620	O,IP	27.0	102

P = Paraffins N = Napthenes A = Aromatics
O = Olefins IP = Isoparaffins

Table 3 – **Fuels Used to Determine Influence of Vaporizing Characteristics**

Cetane Number	Volatility (Final Boiling Point), F	General Composition	Olefins (Bromine No.)	Aromatics (Specific Dispersion)
51	542	P,N,A	2.4	114
51	638	P,N,A	2.1	122
47	686	P,N,A,O	18.6	118
50	726	P,N,A,O	8.0	118

P = Paraffins N = Naphthenes
O = Olefins A = Aromatics

Besides knowing the estimated general composition of the fuels, the bromine number was obtained as a measure of the olefin content and the specific dispersion values[2] used as the criterion for the content of aromatics.

■ Cetane Group

Table 2 shows the first group of five fuels investigated for the effect of the cetane number.

To isolate at least one of the three variables as much as possible, these fuels were selected for constant vaporizing characteristics. The five fuels included a spread of only 39 F in their final boiling points.

A graphical representation of the full-load results obtained with these fuels is given in Fig. 4.

Since determination of the combustion speed was one of the main objectives of this investigation, the abscissa shows elapsed time in milliseconds before and after the top dead center.

The thin lines converging toward the top of the diagram indicate equal crank angle and the duration of the injection is outlined by the cross-sectioned area.

A study of this diagram reveals, first, that the ignition delay periods extend over approximately the same crank angle at all four engine speeds. In other words, less time is required to ignite and burn the fuel at high engine speeds.

[2] See *Industrial & Engineering Chemistry*, Vol. 29, March, 1937, pp. 319-325: "Effect of Solvent Extraction on Aromaticity of Lubricating Oils," by G. H. von Fuchs and A. P. Anderson.

The fuels of different cetane rating show ignition delay in their respective order at the various speeds with exception of the 46-cetane fuel whose curve coincides with that of the 51-cetane fuel at all but the top speed.

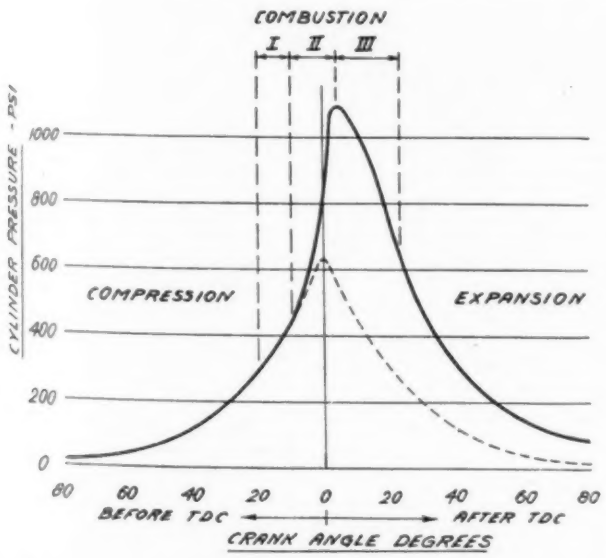
At the 1600 rpm point all three fuels of 57, 51, and 46 cetane show identical ignition delay.

Scanning the tabulated fuel properties for a possible explanation of this behavior, it is seen that neither the boiling range nor the olefin content of these three fuels is materially different. The only distinguishing factor is the materially higher content of aromatics of the 51- and 46-cetane fuels.

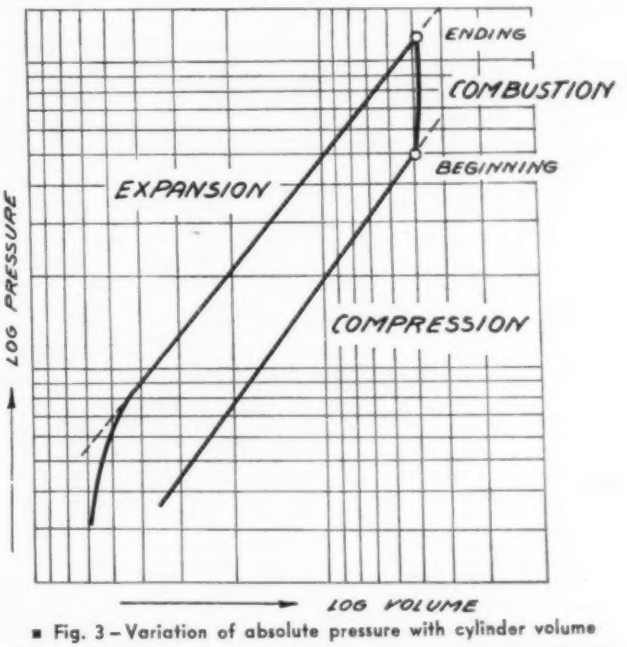
Considering the next phase of the combustion, it is found that, in spite of the relatively large differences in ignition delay, only the 22-cetane fuel shows an appreciable effect of its later ignition. This fuel reaches maximum cylinder pressure much earlier than the other four and is, as a matter of fact, the only one showing the pressure peak before completion of the fuel injection.

Although the 51-cetane fuel has a tendency to burn slowest during this period, the differences between the four fuels appear to be too small to indicate any major effect of cetane rating above 36 on the time of maximum pressure development.

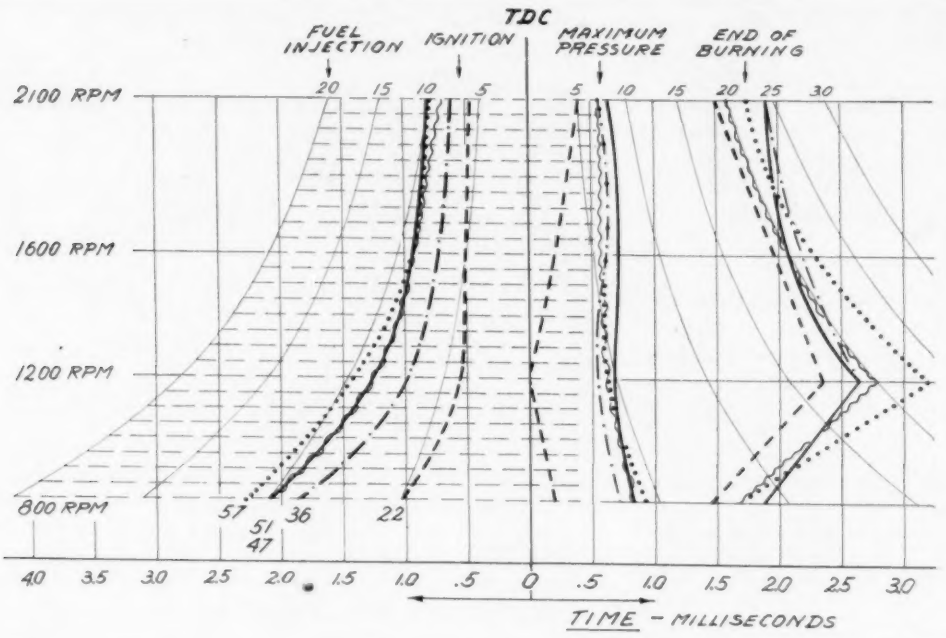
The final combustion period shows several interesting points. In general, the higher cetane fuels have a longer burning period. Both the 36- and 46-cetane fuels burn faster than even the 57-cetane fuel at 2100 and 800 rpm, falling back to their respective places at the 1200 rpm speed.



■ Fig. 2 – Graph showing how fuel heat content is transformed into mechanical power



■ Fig. 3 – Variation of absolute pressure with cylinder volume



■ Fig. 4 – Full-load results – cetane group

The much earlier termination of the combustion at the lowest speed of 800 rpm is due to an air deficiency of the engine for the high fuel input used during these tests. Later work at reduced fuel delivery proved these curves to continue normally in the direction given by the three higher speeds indicating the same time interval for normal combustion.

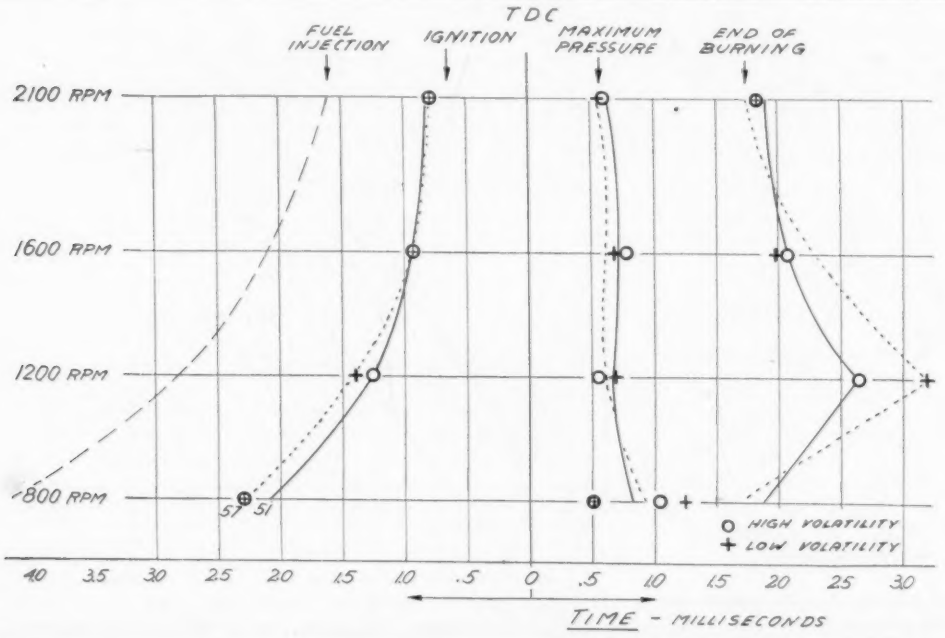
■ Volatility Group

For investigation of the influence of the vaporizing characteristics, a group of four fuels was chosen whose properties are shown in Table 3.

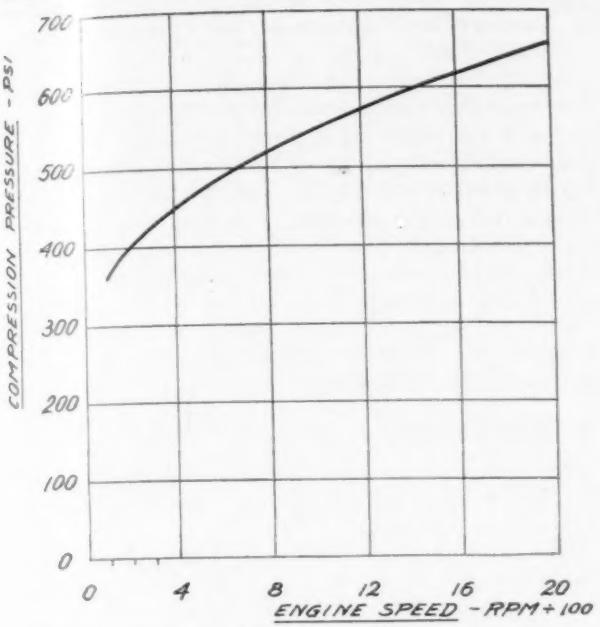
The ignitability of these four fuels varies within four cetane numbers, while their final boiling points differ as much as 184 F. Their aromatics content is in the same order but their olefin content covers a wide range. Fig. 5 contains the results obtained with these fuels added to those of the cetane group.

The ignition delay of all fuels of the volatility group is practically alike and well within the range of the corresponding fuels of the cetane group. Maximum cylinder pressure is reached at approximately the same time as before, and the location of the volatility group fuels at the end of the final burning period indicates likewise that no important influence of the boiling range appears to exist on the pressure development during combustion at full load.

Identical results were obtained in the part-load range



■ Fig. 5 – Full-load results – cetane and volatility groups



■ Fig. 6 – Decrease of compression pressure with speed

down to 50% load, and for all fuels tested neither power output nor fuel consumption showed greater variations than those normally traceable to the slightly different fuel input due to the influence of viscosity and gravity and the differences in heating value.

It must, therefore, be concluded that as far as combustion, power, and economy are concerned, fuel variations within a wide range of properties have no significant effect on this type of a direct-injection engine over the normal range of working loads and speeds.

■ Starting and Light-Load Operation

While high-speed diesel engines at normal operational speeds and reasonable load factors present ideal combustion conditions, thus tolerating a wide variety of fuels with apparently equal satisfaction, the wide speed and load range required of automotive engines creates operating conditions which influence the choice of fuels that will give the greatest operational durability.

Not only is the engine expected to start without undue preparations even after prolonged exposure to low atmospheric temperatures, but light-load operation, including the idling range, is also a necessity for many types of automotive service.

Under these conditions, the identical fuel properties are involved as at full load, but their importance is magnified by factors adverse to vaporization and self-ignition.

Fig. 6 shows the variation of motoring compression pressure as a function of engine speed throughout the normal running range. Comparable curves for various compression ratios have also been published by Heldt.[3]

The instantaneous compression temperatures vary in a similar manner. Under actual firing conditions, however, the temperatures of the combustion-chamber walls are considerably higher and increase with both speed and load.

This increase of the combustion-chamber temperature tends to compensate for the decreasing time interval available for injection, vaporization, and combustion, as shown by Figs. 4 and 5, thus resulting in a combustion process of practically constant angular interval throughout the entire speed range. Furthermore, there is no indication that the speed of compression-ignition engines is limited by the combustion process.

Fig. 7 shows the influence of engine speed on the ignition delay time for the five fuels of the cetane group previously described. To maintain the same time interval for self-ignition at 800 rpm as produced by the 22-cetane fuel at 1400 rpm, an increase of the ignition quality to 57 cetane would be indicated.

Plotting the corresponding values as cetane requirement versus engine speed results in the curve contained in Fig. 8.

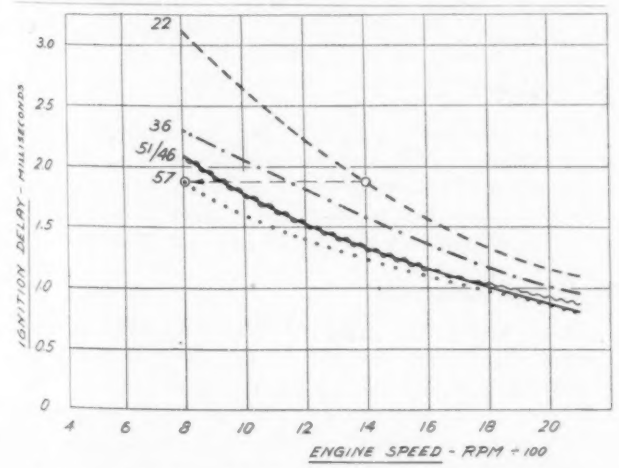
With such a wide variation of the engine ignition capacity over the speed range, it becomes understandable that automotive types of service, such as city coach operation, where low load factors are the rule, make it desirable to employ special fuels.

Practical experience has shown that for operations of this type, volatility and cetane rating have similar effects on overall engine operation, and a deficiency of one can be offset by an increase of the other.

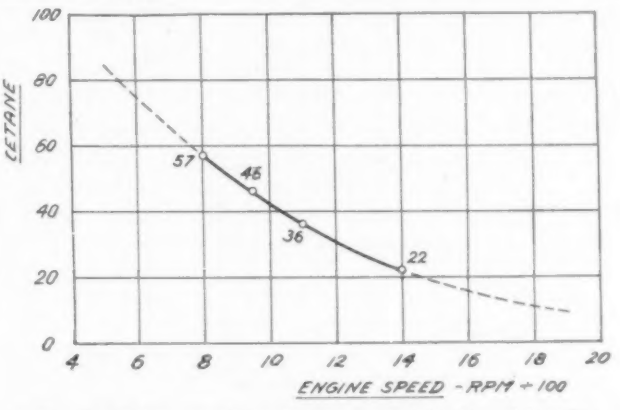
For this reason, the Detroit Diesel Engine Division has advised operators of their engines for several years to select fuels in accordance with the chart shown as Fig. 9.

This chart was originally based on the properties of a

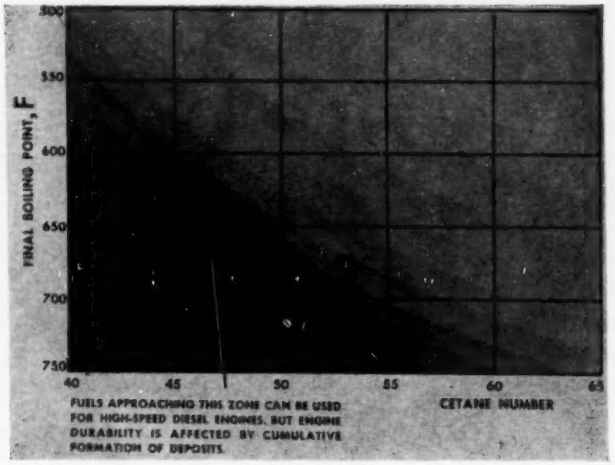
[3] See p. 385 of "High-Speed Diesel Engines," by P. M. Heldt. Fourth Edition, 1943. Published by P. M. Heldt, Nyack, N. Y.



■ Fig. 7 – Variation of ignition delay time with engine speed



■ Fig. 8 – Cetane requirements for constant ignition delay time



■ Fig. 9 – Influence of ignition quality and volatility on engine durability

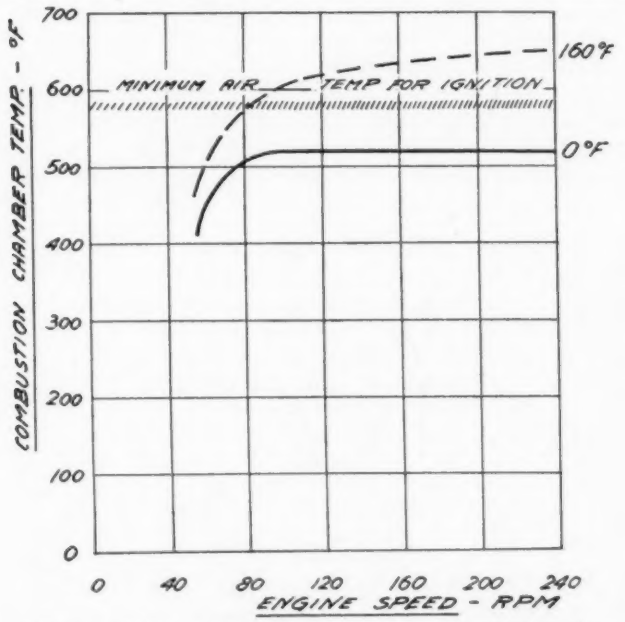
number of fuels which were reported satisfactory from actual field operations. It attempts to demonstrate, by the different intensity of shading, the influence of both volatility and ignition quality on engine deposits and exhaust smoke.

Western stove oils, for example, with usually less than 40 cetane but final boiling points as low as 560 F gave equal operating satisfaction as No. 3 fuel oils with 675 to 700 F final boiling point and 50 to 55 cetane rating.

For city service where drastic restrictions are usually imposed on exhaust smoke and odor, most operators have found fuels preferable with a final boiling point of not more than 530 F and better than 50 cetane number.

Superior diesel-engine fuels must possess both of these important combustion qualities, as indicated by the unshaded area of the chart.

This type of a fuel specification has alleviated a great many misunderstandings on the part of operators and field personnel.



■ Fig. 10 – Effect of intake air temperature on combustion-chamber temperature

Additional information proving this basic interrelation of ignitability and volatility has been provided by more recent experience.

During the war period, tests were conducted for the Army Air Forces with a mixture of 75% aviation gasoline and 25% lubricating oil as emergency fuel for diesel generator sets. The volatility of this mixture was very high but its cetane rating only 21. Both the CFR engine operator as well as the personnel conducting the tests on the multicylinder engine reported that unusually smooth combustion was obtained for a fuel of such a low ignition quality. A subsequent determination of the combustion data of this mixture revealed identical ignition lag with a 21-cetane fuel of natural composition but a 42% longer duration of the combustion.

■ Cold Starting

It is thus seen that frequent operation at light loads and low engine speeds increases the fuel requirements of volatility and cetane rating.

These fuel properties become even more important when the engine must be started in cold weather, and operated with low cooling water and intake air temperatures.

Starting conditions at zero degrees result in combustion-chamber temperatures as shown in Fig. 10. Even continued cranking above 100 rpm does not suffice to raise the air temperature to the level required for ignition of the fuel as determined by the normal starting conditions with 160 F air temperature shown by the upper curve.

A number of carefully correlated cold-starting tests by three different laboratories has produced the relation between cetane rating and starting temperature shown in Fig. 11.

According to these data, fuel of approximately 80 cetane would be required to start a compression-ignition engine without special preheating equipment at an ambient temperature of 0 F. Tests recently conducted with diethyl ether, either as a direct addition to the fuel or separately injected into the intake air, have produced instant cold starts at temperatures as low as —20 F with fuels in the 50-cetane range.

In the absence of any other explanation, it is inferred that the lower starting temperature is the result of the high volatility of the ether since its cetane rating as ordinarily determined in a hot engine is only around 50.

This unusual behavior of ether emphasizes the importance of rating the ignition quality of fuels also in the low-temperature range where ignition is equally important.

■ Engine Life

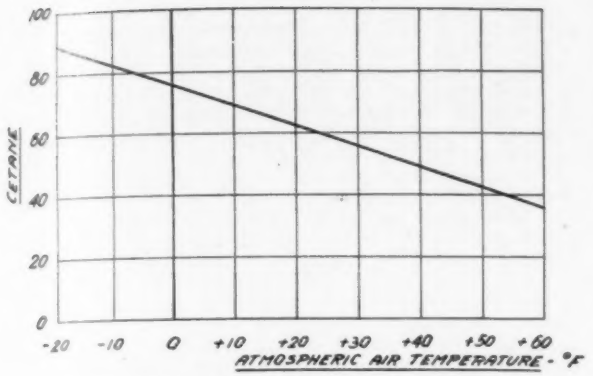
Many diesel-engine operators have reached the important conclusion that low-quality, low-cost fuels are not necessarily the cheapest in the long run. The operational life of an engine may be shortened as much as 50% by use of fuels unsuitable for the type of service encountered.

■ Engine Deposits

Present engine life between overhauls is not limited by normal wear factors but by accumulation of deposits originating from the fuel and the lubricating oil.

As long as the ignitability of the fuel is high enough to prevent actual misfiring, volatility is again the foremost factor.

An excellent criterion of the fuel volatility is presented by



■ Fig. 11 – Cetane requirements for cold starting

the distillation recovery. High-speed diesel engines require first and above all completely distilled fuels, and the fact that they will also run on residual fuel oils has caused heavy repair bills for many operators.

But even if at least a 98% distillation recovery, low Conradson carbon residue, and a reasonably light color indicate completely distilled fuel, the boiling range still represents an important engine life factor.

Besides the unavoidable physical combustion residue which may be minimized by limiting the Conradson carbon content as determined on the last 10% of the distillation to approximately 0.15%, there still appear in the most perfect diesel combustion process small amounts of dry soot originating either from the chilling of some of the burning mixture by contact with cool metal walls or by arresting the combustion by lack of air in the piston-ring zone.

The same effect is obtained when overloading the engine, and the normally small percentage of incompletely burned fuel may then reach considerable proportions.

An excellent grade meter for this type of combustion residue consists of a carefully conducted analysis of the lubricating oil for combustible solids.

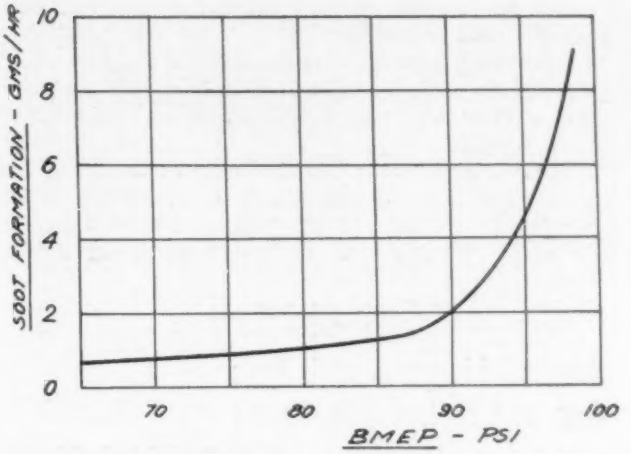
Arter[4] has published the information graphically shown in Fig. 12.

At approximately 90 bmep there appears a decided break in the oil contamination curve, and although the soot formation at this point amounts to only 0.02% of the fuel consumed, for a 150-hp engine burning 9.5 gal of fuel per hr there remain 13.5 lb of carbon which have to pass the piston rings and skirts every one thousand hours of operation. On their way down to the crankcase some of these carbon particles combine with partly oxidized lubricating oil to form the obnoxious sticky deposits well known to every diesel-engine operator.

Oil changes and filter life depend largely on the amount of this contamination which appears, as long as the normal load is not exceeded, to be directly proportional to the final boiling point of the fuel.

For best life of high-speed engines, a high limit of 675 F for the final boiling point should not be exceeded.

To conclude the list of fuel properties which limit engine life, mention must be made of the sulfur content of fuel oils, which has lately been brought out into the open as a potential contributor to engine deposits by the excellent



■ Fig. 12 – Effect of engine load on rate of soot formation

investigations of Cloud and Blackwood of the Standard Oil Development Co.[5]

In this case, the sulfur appears to be harmful not in its original form, but the trioxide formed during the combustion combines with the lubricating oil to insoluble sludges, lacquers, and hard carbon deposits.

Although increasing use of additive oils of the heavy-duty type appears to minimize deposits from this source, a sulfur limitation well below the present maximum of 1% seems advisable.

Further help may be expected from the fact that the new catalytic cracking processes produce fuel oils which are essentially low in sulfur content.

■ Conclusion and Acknowledgment

This discussion covers a portion of the data accumulated in a study of fuels and combustion made by the Detroit Diesel Engine Division.

For the type of engine under investigation the increase in temperature and pressure with increasing load and speed is such that for any given fuel the angular interval measured in crankshaft degrees from the beginning of injection to the end of combustion is practically constant for all speeds over the normal operating range.

A wide variation of those fuel properties which are normally associated with the burning characteristics of diesel fuels is of insignificant influence on the combustion of the General Motors two-cycle, direct-injection engine under normal load and speed conditions.

The power developed in a diesel-engine appears to be directly proportional to the heating value of the fuel injected and is independent of its cetane rating, volatility, chemical composition, and molecular structure.

Both the cetane rating and the volatility have a marked effect on cold starting and light load operation, when the ignition capacity in the cylinder is reduced.

One of the main factors governing the requirements of diesel-engine fuels is the type of service in which the engine is operated. For automotive engines for highway vehicles and other similar services the engine must not only run well at high speed and maximum power, but also perform satisfactorily under operating conditions involving light engine loads and low speeds, and particularly cold-starting conditions.

These operating conditions must, therefore, be consid-

[4] See *Institution of Automobile Engineers Journal*, Vol. 12, December, 1943, pp. 13-30: "Heavy-Duty Motor Oils," by K. T. Arter.

[5] See SAE Journal (Transactions), Vol. 51, November, 1943, pp. 408-414, 419-420: "Influence of Diesel Fuel Properties on Engine Deposits and Wear," by G. H. Cloud and A. J. Blackwood.

ered as the criterion by which automotive diesel fuel specifications must be established.

Acknowledgment is due to the Shell Oil Co., Inc., which supplied the majority of the test fuels and their analysis data together with the valuable advice and assistance of L. A. Wendt.

The combustion tests described were conducted by Dr. L. L. Withrow and the late Dr. A. S. Fry of the Research Laboratories Division of the General Motors Corp. in the engineering laboratory of the Detroit Diesel Engine Division.

DISCUSSION

Substantial Improvement in Fuels Seen as Still Needed for Diesels

—HAMPTON H. FOSTER

National Advisory Committee for Aeronautics

THIS writer does not share the authors' enthusiasm expressed in the introduction of their paper regarding the implied great improvements in diesel fuels and the advancement of the diesel engine as a result of such improvements. True, there have been great improvements in spark-ignition engine fuels, where the problem was fuel knock rather than fuel mixing with the combustion air.

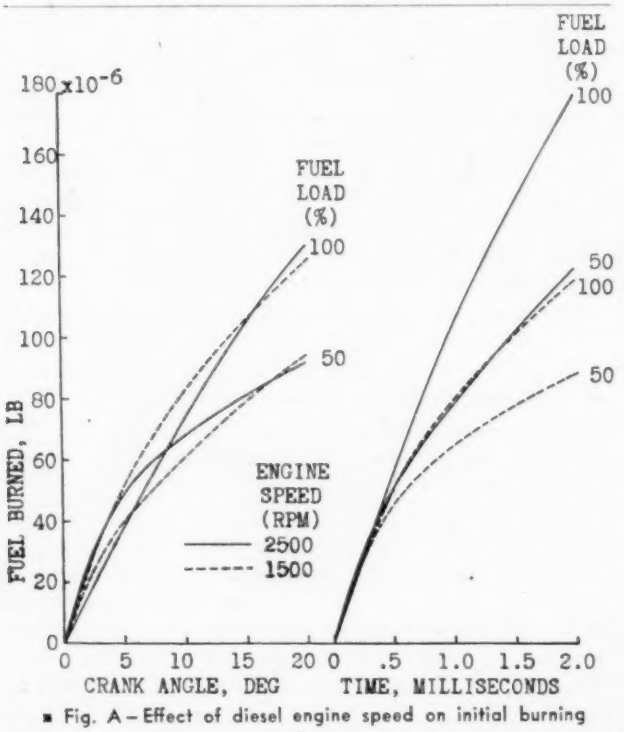
From the viewpoint of diesel-engine starting reliability and maintenance, progress has been made as a result of better diesel fuels; but from the viewpoint of increased brake mean effective pressures and fuel economy, little progress has been made for years. Such an improvement could result from a fuel of higher heat content or a fuel that could be more readily mixed with the combustion air.

The writer would like to offer supporting data on two items in the paper, namely:

(a) The use of gasoline-oil mixtures as diesel fuels.

(b) The effect of engine speed on combustion.

Gasoline-Fuel-Oil Mixture—It was thought that a mixture of 25% gasoline and 75% fuel oil would evaporate faster than fuel oil alone, and that more of the total fuel charge would burn earlier than for the plain fuel oil. The fuel oil was expected to ignite first, owing to its lower ignition temperature, but the gasoline was expected to evaporate more quickly than the fuel oil and to produce better combustion during the early part of the expansion stroke.



■ Fig. A—Effect of diesel engine speed on initial burning

The cetane number of the mixture was 42.4, compared with 62.5 for the fuel oil alone, and the ignition lag in the engine increased 17.5%, possibly because of the cooling action of the rapidly evaporating gasoline. The rate of pressure rise decreased 14%, the maximum cylinder pressure decreased from 920 to 875 psi, and the engine operation was considerably smoother and quieter despite the much lower cetane number. In this instance, cetane number falls down as a fuel-rating method.

The engine-performance test of the gasoline-fuel-air mixture showed that the mean effective pressure was 3% lower than for the fuel oil alone, and that the fuel consumption increased. As the heating value of the fuel mixture was the same as for a like weight of fuel oil, the effect of the gasoline must have been to cause slower rather than faster burning, with a consequent decrease in power.

The addition of 25% gasoline to fuel oil greatly increased the fire hazard, because it decreased the flash point from 236 to below 80 F.

Effect of Engine Speed on Combustion—A plot of total fuel burned after ignition with respect to crank angle and also with respect to time in milliseconds for the initial combustion in a diesel cylinder is shown in Fig. A for engine speeds of 1500 and 2500 rpm, for full- and half-load fuel quantities.

The data for this plot were obtained from an analysis of pressure-time records from a displacer-piston type, four-stroke cycle, 5 by 6, single-cylinder test engine.

The zero point on the abscissa is not top center, but is made the common starting point of the curves in order to obtain a comparison of the combustion progress in the early part of the power stroke. In the right side of the figure, 2 milliseconds is equivalent to 18 deg crank angle at 1500 rpm, and 30 deg crank angle at 2500 rpm.

The higher combustion speed at the higher engine speed becomes apparent when the abscissa of the plot is time rather than crank degrees. This trend is in agreement with the results obtained by the authors and may be considered a natural result of the increase in turbulence, pressure, and temperature in the combustion chamber at the higher engine speed.

The large increase in rate of combustion with increase in engine speed may come somewhat as a surprise in that there seems to be a general belief that the rate of combustion is more or less fixed and would, therefore, prove to be a limiting factor in high speeds for diesel engines. An example of speeds that have been attained, even 10 years ago, is the British Perkins Vixen automotive diesel engine, which is listed as having a speed range of from 300 to 4000 rpm.

TRANSMISSIONS

continued from page 338

Assuming equally good design, the average efficiency of a differential electric transmission naturally is higher than that of a straight electric drive, because with the former only a fraction of the total power is subjected to a double conversion, the remainder being transmitted directly. In the bus installation the transmission had an equilibrium speed of about 14 mph, and at this speed all of the power received by the driving wheels was transmitted directly. Below this speed the percentage of power converted increased rapidly, to 100% at 7 mph, while above the equilibrium speed it increased more gradually, to about 30% at 35 mph.

The differential principle can be applied also to hydrokinetic torque converters.

Summarizing, there are at least four different types of drive or transmission with which uninterrupted acceleration can be obtained, namely, the straight electric drive, the hydromechanical transmission, the combined hydrokinetic torque converter and coupling, and the differential transmission. Of these the straight electric drive has been in use longest. Two large fleets of city buses were equipped with it as far back as 1925. During the past decade extensive use has been made of the hydromechanical transmission. So far as I know, neither the combined converter and coupling nor the differential transmission has been used in the bus field, except in an experimental way, but it is my opinion that both possess considerable merit and that much will be heard of them in the future.

Factors Affecting the Design of JET TURBINES

by **WILLIAM R. HAWTHORNE**
British Ministry of Aircraft Production

GREAT engineering resources have been expended on the development of the reciprocating engine as an aircraft powerplant. Extensive researches have been made into detonation. Progressive improvements have been made in materials and processes of manufacture. Most of the skill and patience of the engineer has, however, been devoted to solving the mechanical problems of the engine. In fact, it now takes several years for a modern aircraft engine to reach a state of maturity, and most of this period is spent running the engine, or parts of it, on the test bed to find and overcome mechanical weaknesses, which cannot be predicted in the design stage. Consequently, aircraft-engine design is an art, the main prerequisite for which is an aptitude for mechanical design.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 9, 1946.]

THE advent of the gas turbine has changed the character of the aircraft-engine designers' problems. The use of rotating machinery and continuous flow simplifies the mechanical design and the prediction of performance.

To illustrate this the author has analyzed the factors affecting the thrust output of a Whittle jet turbine. These show that the design is a compromise between aerodynamic and stress limitations. At the moment the metallurgist appears to have contributed more to the success of the gas turbine than the aerodynamicist, although present achievements have been made possible by the work of both. Future progress, the author feels, will depend on continuous research in both fields.

In making the conversion from the piston engine to the gas turbine, he concludes, the design office will have to furnish itself with some new equipment, the major item will be a fund of aerodynamic knowledge.

THE AUTHOR: WILLIAM R. HAWTHORNE, deputy director in charge of engine research at the British Ministry of Aircraft Production, has filled the 12 years since his graduation from Cambridge University with a variety of interesting activities. After a year with Babcock & Wilcox, Ltd., he spent two years as a post-graduate at M. I. T. When the war broke out in 1939, he joined the Royal Aircraft Establishment, was loaned to Power Jets, Ltd., to do their combustion chamber development, spent three years in charge of turbine problems, and was later transferred to the British Air Commission.

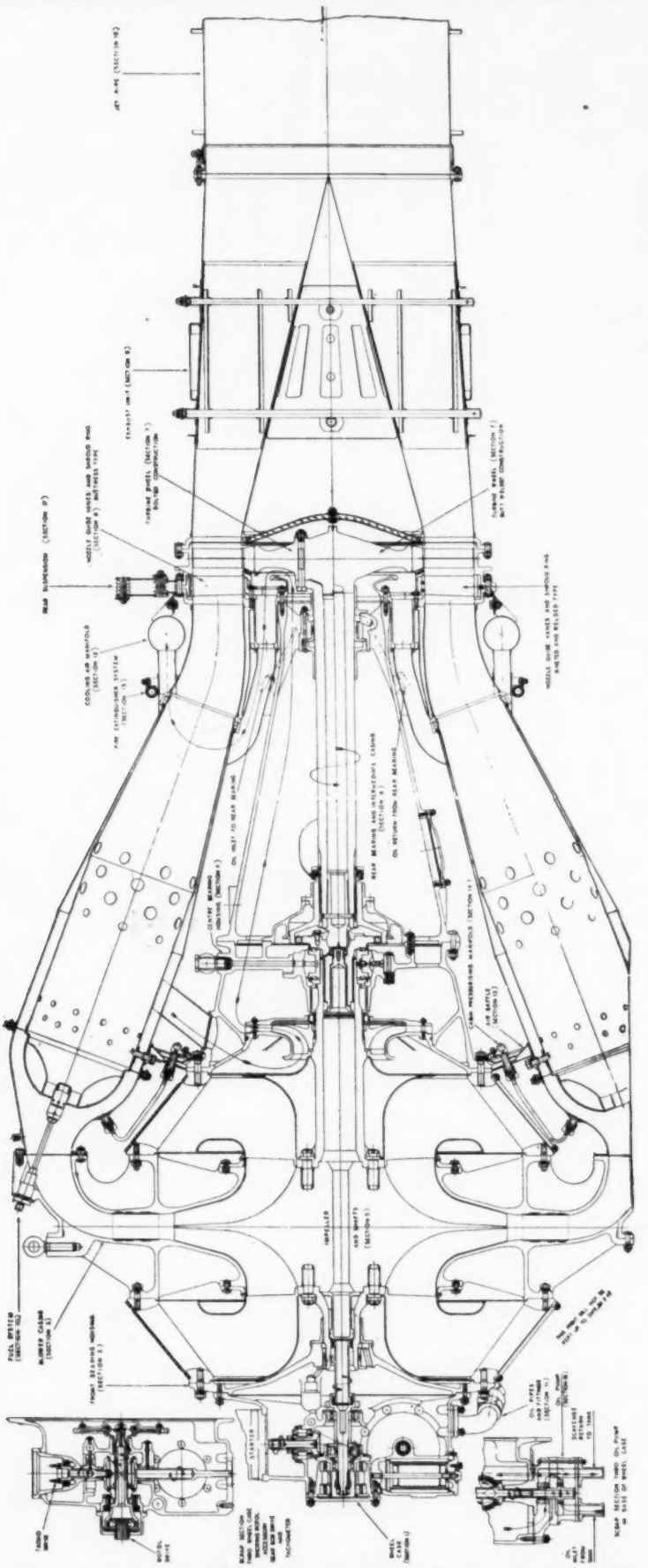
The gas turbine which is now taking the place of the piston engine for aircraft propulsion presents problems of a different nature. While its mechanical problems are not to be ignored, the stressing problems are simpler and the aerodynamic problems assume a major importance. The continual development of materials, particularly for turbine wheels and buckets, is essential. The combustion problems, which are largely aerodynamic, involve much research. The aircraft-engine designer of the future will be preoccupied less with problems of mechanical design and more with problems of fluid motion than his predecessor.

Because of the relative simplicity of rotating machinery and of continuous flow processes, the performance of turbine engines can be predicted with greater confidence than has ever been possible with reciprocating engines. This simplicity is enhanced by the functional separation of the components, compression, combustion, and expansion occurring in different parts of the engine instead of in a cylinder. Each component may, therefore, be put through a separate process of aerodynamic development on suitable test rigs. Using the results of such tests and his experience of aerodynamic problems, the designer may scheme different arrangements of the various components and assess their merits with fair confidence in his calculations. Consequently, while the gas turbine offers a challenge to the aerodynamic skill of the designer, its advent permits him to employ the powerful weapon of analytical calculation to an extent hitherto impracticable. Experience has already shown the value of the analytical approach in the design stage, in predicting possible troubles, and in the interpretation of test results.

This paper is presented as an illustration of the analytical approach. It reviews the problems of designing a jet turbine with a centrifugal compressor, most emphasis being laid on the aerodynamic and stress limitations.

■ Major Problems

An arrangement of jet turbine is shown in Fig. 1. It consists of a centrifugal compressor, in this instance of the double-entry type, followed by a number of pipe-type combustion chambers, a single-stage axial-flow turbine directly connected to the compressor, and an exhaust duct. The propelling nozzle is fitted at the end of the exhaust duct. The history of the early development of this general type



■ Fig. 1 – General arrangement of jet turbine with centrifugal compressor

of gas turbine has been described by Whittle, its inventor.[1] One of its major advantages over the reciprocating engine and propeller is its high power-weight ratio. Simple calculations, based on the thermodynamic cycle used, show that the thrust obtainable per unit rate of airflow is a function of the compressor pressure ratio, the temperature at the inlet to the turbine, and the efficiencies of the components. The pressure ratio and the turbine inlet temperature for which the engine will be designed are chosen from considerations of the strength of the material available to make the compressor and turbine rotors and the turbine buckets. Once these have been chosen the thrust obtainable from a given size of engine will depend on the amount of air which can be passed through it and on the efficiencies of the components.

Consider for a moment a duct through which air is passing. The maximum flow rate for a given inlet pressure and temperature is achieved when the velocity of flow over some cross-section transverse to the principal direction of flow reaches the velocity of sound. The velocity of sound depends only on the temperature of a given gas and hence varies somewhat through a system such as a gas turbine. It is, therefore, more convenient to speak of the Mach number, which is defined as the ratio of the local velocity of flow to the local velocity of sound. Hence, the attainment of a Mach number of unity at any cross-section of the duct results in a limitation to the flow rate. The mass flow rate may then only be raised by increasing the pressure or reducing the temperature of the air entering the duct. The relation between mass rate of flow, Mach number, pressure, and temperature is derived in the Appendix.

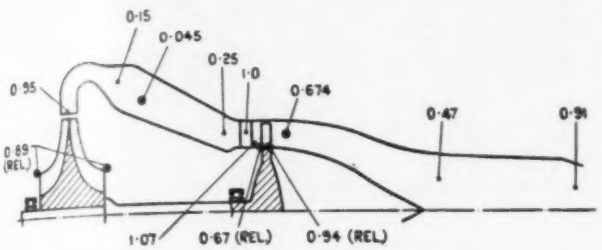
The Mach number is a useful parameter not only because it defines the limiting or "choking" flow but also because aerodynamic losses are often increased as the Mach number approaches unity. The classical example is the aircraft wing whose drag at high speeds is a function of Mach number, the drag coefficient increasing rapidly as the Mach number of the air approaching the wing is raised above a critical value of the order of 0.7 to 0.8. The effect of increasing Mach number on loss is not so noticeable where the flow is accelerating, as in turbine nozzles and buckets, but its influence on the efficiency of compressors and diffusers is considerable.

As the Mach number is such an important parameter in the gas turbine, its values at various points through an engine of the type given in Fig. 1 are shown in Fig. 2. The Mach numbers are quoted for the design point, which will be the only engine condition discussed in this paper and is taken as the sea level, static (0 mph) condition and maximum rotor rpm. The atmospheric conditions correspond to air at 14.7 psi, absolute pressure and 520 R (60 F). In Fig. 2 values of the Mach number at entry to the impeller of the compressor and in the turbine buckets are quoted for the velocities relative to the moving vanes or buckets.

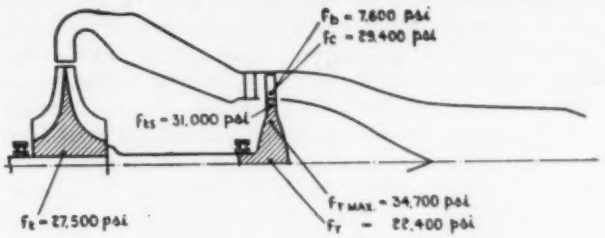
At the ringed points in Fig. 2 the choice of Mach number to be used for design purposes may greatly influence the size of the engine. The points are:

1. The tip of the rotating guide vanes at the inlet to the compressor.

[1] "Early History of Whittle Jet Propulsion Gas Turbine," by F. Whittle. Presented at a meeting of the Institution of Mechanical Engineers, London, England, Oct. 5, 1945.



■ Fig. 2 – Values of Mach number at various points in jet turbine for designed rpm, sea level, 0 mph conditions



■ Fig. 3 – Principal stresses in rotating parts of jet turbine for designed rpm, sea level, 0 mph conditions

f_b = Gas bending stress on turbine bucket
f_c = Centrifugal stress at bucket root
f_{rs} = Radial stress in disc rim between bucket root fixings
$f_{r(max)}$ = Maximum radial stress in disc
f_r = Radial stress at centre of disc
f_t = Hoop stress at impeller bore

2. The inlet to the combustion chamber.
3. In the turbine buckets.
4. The exhaust annulus at the outlet from the turbine.

The effect of choosing different Mach numbers at these points will be analyzed later.

The choice of working pressure ratio and turbine inlet temperature depends largely on the stresses which the rotating parts of the engine will stand. To achieve a high pressure ratio from the compressor at high tip speed is required, and high working stresses in the impeller consequently result. To achieve a high work output from the turbine a high rim speed is necessary and the stresses in the disc are, therefore, high. The principal stresses in the moving parts of a typical engine are shown in Fig. 3.

More gas can be passed through the turbine buckets if they are lengthened, but lengthening them increases the centrifugal stress at the root of the bucket and in the rim of the disc adjacent to the root fixing. With the "fir tree" type of root fixing described by Whittle[1] these two stresses are of the same order. It will be shown in the later analysis that the value selected for the centrifugal stress in the turbine buckets may have an important effect on the output of the engine.

A major problem with the jet turbine is its fuel consumption. At first sight it appears to be appreciably greater than that of a piston engine, but it is to be remembered that the two engines are not in the same class for power-weight ratio. Nevertheless, great attention to design is required to reduce the fuel consumption. High component efficiencies and low aerodynamic losses are necessary. The effect of increasing the Mach numbers in, and the air swallowing capacities of, the various components has been noted. At certain critical points, such as the entry to the impeller, the choice of Mach number may, therefore, be a compromise between efficiency and output.

The principal components of the engine will now be discussed in more detail.

■ Compressor

The problems of the centrifugal compressor are already well-known to the designer of piston engines, for it is this component which is common to both the piston engine and the Whittle type of gas turbine. In the gas turbine, however, both the size and the impeller tip speed have been increased beyond the values generally used in the supercharger. The increase in size, together with some intensive development work, has resulted in efficiencies in the range from 71% to 75% being attained at tip speeds of 1500 fps. The pressure ratios achieved are consistent with these efficiencies. Data from tests of a typical jet turbine are given in Fig. 4 in the form of a plot of pressure ratio versus tip speed. The slip factor, defined as the ratio between the tangential velocity of the air leaving the impeller tip and the tip speed, varies between 0.89 and 0.91.

While the details of impeller and diffuser design considerably affect the efficiency and characteristics of the compressor, the air swallowing capacity is chiefly determined by conditions at the inlet to the impeller. For, although the flow at the entry to the diffuser vanes may reach the velocity of sound, the designer can select suitable diffuser throat areas to pass the flow required without influencing the overall size of the compressor. The flow through the inlet or impeller eye may be increased until the Mach number relative to the tip of the rotating vanes reaches such a value that, due to compressibility effects, a stall occurs and the compressor efficiency begins seriously to deteriorate. Assuming the absolute velocity of the flow approaching the eye is uniform and in an axial direction, a theoretical expression has been derived (see Appendix) relating the mass flow rate and the impeller tip, eye, and hub dimensions to the Mach number and the tip speed. This relationship is shown in Fig. 5. The curves show that the mass flow rate W through the eye of the impeller is proportional to the square of the impeller tip diameter D. At a given impeller diameter D, tip Mach number M, and tip speed U, the mass flow increases as the eye diameter is increased up to a certain value, beyond which there is a rapid drop. The mass flow increases with the tip Mach number.

The choice of tip Mach number is a compromise. Campbell and Talbert[2] recommend, on the basis of experience with superchargers, that a value of 0.75 should not be exceeded. Careful attention to the inlet vane design and the use of a large number of vanes appear to make possible the attainment of higher values without a serious loss in efficiency. Values of the order of 0.9 from tests of current jet turbines are shown in Fig. 5. It should be noted that when a double-entry impeller of the type shown in Fig. 1 is used, the mass flow through the compressor is double that shown in the curves of Fig. 5.

The maximum mass flow for a given tip speed and Mach number is obtained with a certain optimum eye dimension. If it is assumed that the optimum eye dimension be chosen

[2] See SAE Transactions, Vol. 53, October, 1945, pp. 607-618 + (disc.) 618-620: "Some Advantages and Limitations of Centrifugal and Axial Aircraft Compressors," by K. Campbell and J. E. Talbert.

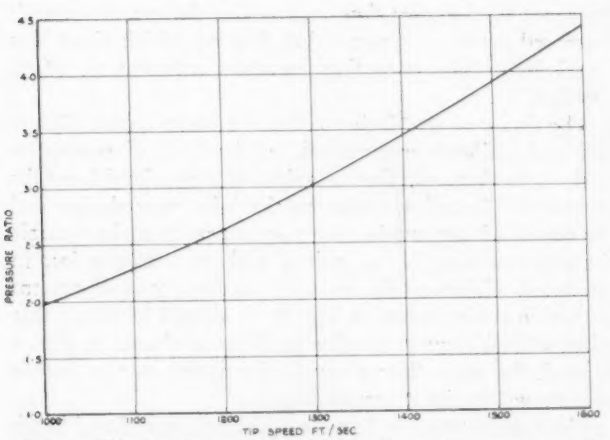
for each condition, a relation for the ratio of maximum mass flow to the square of the tip diameter D^2 may be obtained for each tip speed and Mach number. This relation is shown graphically in Fig. 6, the curves being constructed for an impeller with a hub/tip diameter ratio of 0.3. The curves show that, as the tip speed is increased, there is a decrease in the maximum mass flow which can be passed through the impeller without incurring losses due to a high Mach number at the tip of the eye. Here again the flow rates shown in the figure should be doubled for a double-entry impeller.

The elastic stresses in a typical impeller are shown in Fig. 7 for a tip speed of 1490 fps. The impeller shown has an axial hole bored through the hub to permit inspection and simplify the process of heat-treatment. With materials now available it is possible to use tip speeds of up to 1600 fps for take-off conditions.

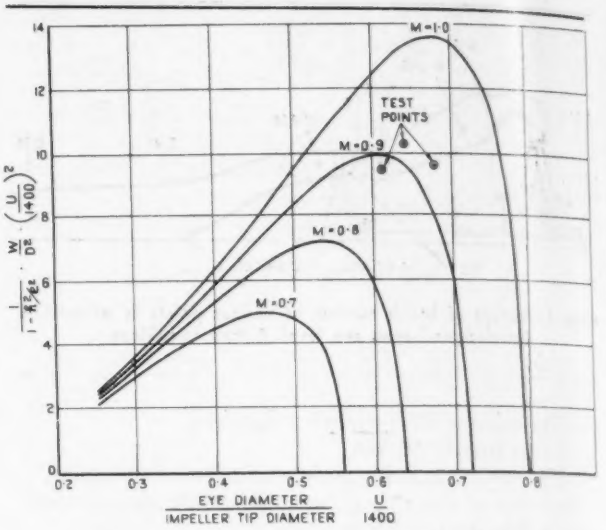
■ Combustion Chamber

In the early development of the jet turbine[1] the combustion chamber was the most troublesome component. The Germans also found it to be the Achilles' heel of the engine. Although great improvements have now been made, the combustion chamber should still receive a greater share of attention from the development team than any other part of the engine. It is beyond the scope of this paper to deal with the many aspects of the problem. The combustion process is extremely sensitive to slight changes in the pattern of flow of air and fuel. The velocity distribution at the entry to the combustion chamber requires careful attention. Meticulous care should be taken to achieve accuracy in the manufacture and assembly of the sheet metal parts and the fuel injectors.

In laying out the combustion chamber the designer should, therefore, ensure that the flow at entry is as uniform as possible. Most experience has been obtained with systems comprising a number of tubular chambers connected in parallel and disposed around the rotor. Ratios of length to maximum diameter have varied between 2¼ and 5. Heat release rates in the volume swept by hot gases of 1.5 to 4 million Btu per cu ft per hr per atmosphere of pressure in the chamber have been used. In order to keep the pressure loss low, low velocities and large cross-sectional areas are required. The ratio of cross-sectional area of the combustion chambers to the overall frontal area of the



■ Fig. 4 – Relation between pressure ratio and compressor tip speed



■ Fig. 5 – Air swallowing capacity of impeller eye

D = Impeller tip diameter, ft
U = Impeller tip speed, fps
W = Mass flow corrected to standard pressure (14.7 psi, absolute) and temperature (520R)
h = Hub/tip diameter ratio
e = Eye/tip diameter ratio
M = Mach number of flow relative to tip of rotating inlet vanes

Note: Figure of 1400 has no physical significance and is chosen merely to give convenient scales

combustion chamber assembly should, therefore, be as large as possible. For tubular chambers disposed around the rotor this ratio is a function of the number of chambers, as shown in Fig. 8. When four chambers are used, the maximum cross-sectional area is obtained and is equal to 62.5% of the total frontal area. The use of only four chambers, however, leads to difficulties in the arrangement of diffusers and discharge ports from the compressor casing and to some cramping of the space available inside the assembly for the rotor and its housings. In British engines 7 to 16 chambers have been used, the larger numbers making possible a decrease in the dimensions of the compressor casing at the expense of some increase in pressure loss.

A limit to the flow through the combustion chamber occurs when the Mach number at the outlet reaches unity, but this has never been approached in gas turbines because of the high pressure losses involved. The pressure loss may conveniently be expressed in terms of the percentage of total pressure at the entry to the combustion chamber. At a pressure ratio of 4 to 1, a combustion pressure loss of 3% is equivalent to about 2% reduction in compressor efficiency. For convenience in comparing the losses obtained in different chambers a combustion-chamber Mach number may be defined as the value associated with the velocity which the entering air would have if it flowed uniformly through the maximum cross-section of the chamber (M_2, see Fig. 9). A good combustion chamber has a percentage loss of the order of 1500 times the square of the combustion-chamber Mach number. Hence, in order not to cause a loss equivalent to more than 2½% decrease in compressor efficiency the combustion-chamber Mach number should not exceed a value of about 0.05.

■ Turbine

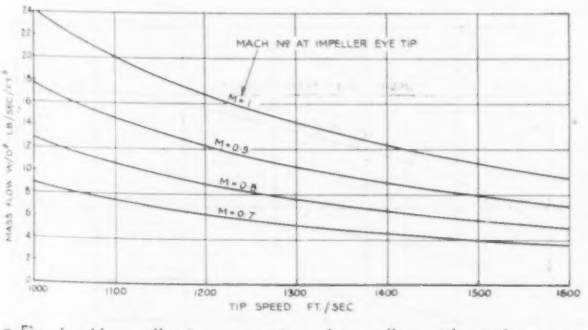
In jet engines the turbine is only required to drive the

compressor plus a few engine auxiliaries and aircraft accessories geared to the rotor. Allowing for this and for windage losses, it is reasonable to assume that the turbine work is between 2 and 3% more than the work done on the air by the compressor. It may also be assumed that the addition of the fuel mass to the working fluid in the combustion chambers is balanced by the removal of air from the compressor for cooling purposes, so that turbine and compressor handle the same mass flow. Allowing for the different specific heats of gas and air, it follows that the temperature drop, or the temperature equivalent of the work required from the turbine, is about 0.9 times the temperature rise in the compressor.

In single-stage turbines for jet engines the velocity of the gas leaving the turbine nozzles is supersonic. A throat, therefore, exists in the turbine nozzles and the mass flow is proportional to its cross-sectional area. The overall dimensions and weight of the turbine required for a given duty do not, however, depend directly on the throat area. The area and the mass flow may, in fact, be increased until the velocity of sound is reached at the outlet from the buckets or in the propelling nozzle. The mass flow will then reach the limiting value for the given inlet pressure and temperature. Because the size of the propelling nozzle may be easily altered, the maximum capacity of a turbine occurs, therefore, when the flow leaving the buckets reaches the velocity of sound. The convenient parameter is the Mach number (M_4, see Fig. 9) based on the axial velocity of the gas in the turbine exhaust annulus immediately downstream from the turbine buckets. It is assumed that the radial height of the exhaust annulus at the point considered is equal to the radial height of the gas swept trailing edges of the turbine buckets. The velocity on which the Mach number is based is the mean value over the annulus area and relative to the fixed walls of the exhaust duct. The choice of turbine exhaust Mach number and annulus area greatly affect the mass flow and thrust output of the engine as the subsequent analysis will show.

In jet turbines with centrifugal compressors the frontal area of the turbine is appreciably less than that of the compressor and combustion-chamber assembly. The size of the turbine disc is, therefore, not a factor which determines the overall dimensions of the engine. Smaller discs, however, lead to lower weights and stresses. The stresses in a disc of given shape are roughly proportional to the square of the rim speed, for the secondary effect of the loading due to the buckets may be neglected in a preliminary analysis. The rim radius is defined as the inner radius of the turbine exhaust annulus.

A lower limit to the rim speed is set by the flow conditions through the root section of the buckets. Here, if a slight pressure drop (or reaction) is to be maintained through the buckets and excessive swirl in the exhaust annulus is to be avoided, the temperature equivalent of the work done may not exceed a certain factor times the square of the rim speed. If the temperature drop is expressed in Fahrenheit degrees and the rim speed in feet per second, this limiting proportionality factor is 4.5×10^{-4}. It is advantageous from the point of view of efficiency to work well within the value of the factor quoted, in other words to use a high rim speed. Also the turbine bucket temperatures may be reduced by increasing the rim speed. This conflicts with the requirement of low rim speed to keep the disc stresses and weight low. In making the compromise, British designers have used values of the rim speed ranging from about 850 to 1000 fps.



■ Fig. 6 – Air swallowing capacity of impeller with optimum eye dimensions

D = Impeller tip diameter, ft
U = Impeller tip speed, fps
W = Mass flow corrected to standard pressure (14.7 psi, absolute) and temperature (520R)
h = Hub/tip diameter ratio = 0.3 for curves shown
M = Mach number of flow relative to tip of rotating inlet vanes

■ Synthesis

The effect on the output and performance of the jet turbine may now be shown in terms of the aerodynamic and stress limitations in its components. In the analytical presentation the subscripts shown in Fig. 9 will be used to define the conditions of the working fluid in the various parts of the engine.

It is convenient to start at the turbine exhaust annulus where, if A is its cross-sectional area (sq ft), the expression for the flow W (lb per sec) is:

$$W = \rho_4 A V_4 \tag{1}$$

where ρ_4 is the density of the gas (lb per cu ft) and V_4 is the axial velocity (fps).

In the Appendix [equation (20)] it has been shown that equation (1) may be written:

$$W = \frac{p_{4t} A}{\sqrt{T_{4t}}} \sqrt{\frac{g\gamma}{R}} \frac{M_4}{\left(1 + \frac{\gamma - 1}{2} M_4^2\right)^{\frac{\gamma + 1}{\gamma - 1}}} \tag{2}$$

thus relating mass flow to Mach number at the turbine exhaust.

Here:

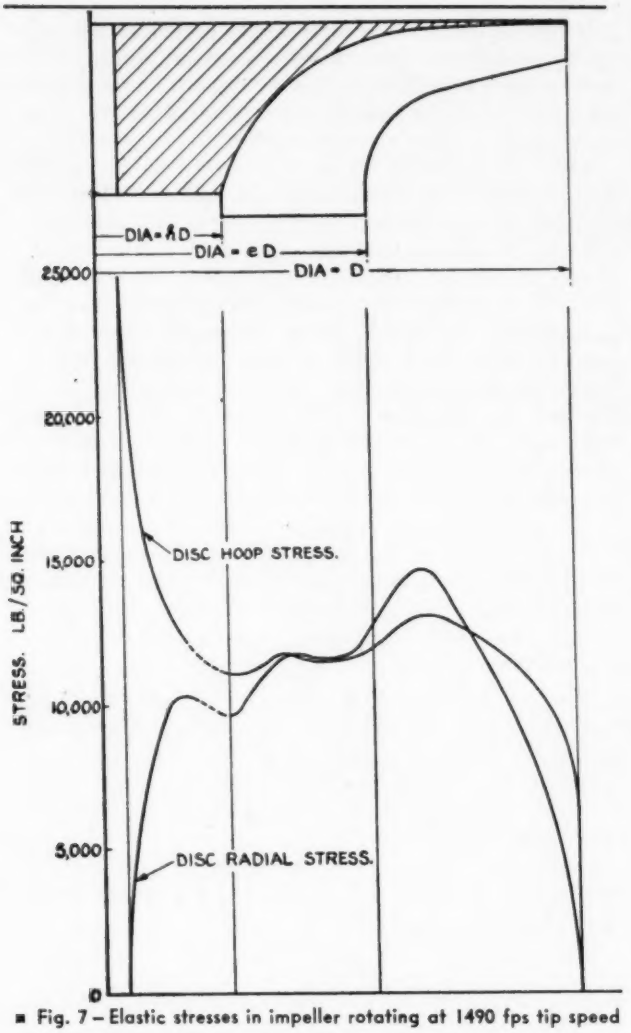
M_4 = Turbine exhaust Mach number
γ = Ratio of specific heats
g = Acceleration due to gravity
R = Gas constant
p = Absolute pressure, psf
T = Absolute temperature, R

Subscript t is used to denote stagnation or total head values.

For the expansion through the turbine:

$$\frac{p_{4t}}{p_{3t}} = \left(1 - \frac{1}{\eta_t} \frac{T_{3t} - T_{4t}}{T_{3t}}\right)^{\frac{\gamma}{\gamma - 1}} \tag{3}$$

where η_t is the turbine efficiency on a total head basis defined as the ratio of the temperature equivalent of the work done in the turbine to the drop of temperature in isentropic expansion from p_{3t} to p_{4t}.

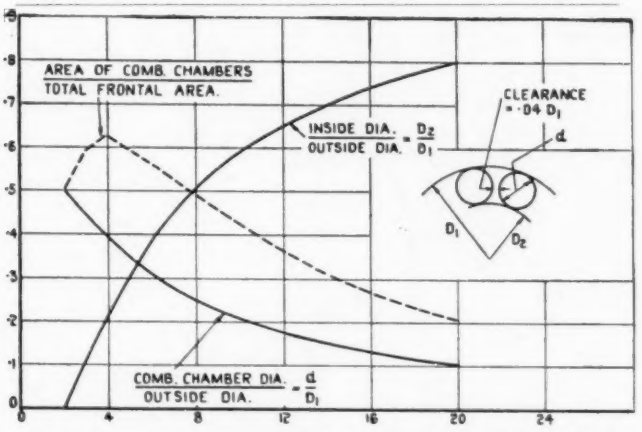


■ Fig. 7 – Elastic stresses in impeller rotating at 1490 fps tip speed

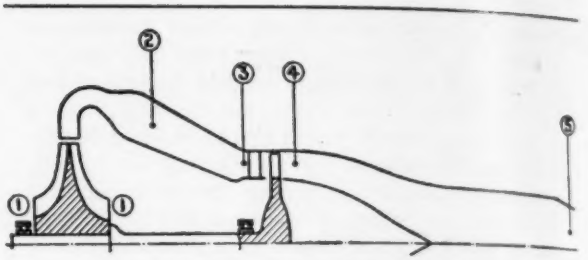
The work in the turbine has to be equated to the work in the compressor, hence:

$$(T_{3t} - T_{4t}) = 0.9\ (T_{2t} - T_{1t}) \tag{4}$$

By inserting values for p_{4t} and T_{4t} obtained from equations (3) and (4) in equation (2) the mass flow per square foot of exhaust annulus area can be obtained in



■ Fig. 8 – Effect of varying number of combustion chambers on total cross-sectional area available to flow – clearance is 4% of outer diameter



■ Fig. 9 – Definition of subscripts used in equations

terms of the pressure p_{3t} and temperature T_{3t} at the inlet to the turbine, the temperature rise or work done in the compressor, $T_{2t} - T_{1t}$, and the turbine exhaust Mach number M_4.

The thrust per square foot of exhaust annulus area may now be obtained by deriving the expression for thrust per unit rate of airflow through the engine.

The thrust is proportional to the product of the mass flow and jet velocity V_5, being given by:

$$\text{Thrust} = \frac{W\ V_5}{g} \tag{5}$$

The jet velocity V_5 is obtained by expansion from p_{4t} and T_{4t} to atmospheric pressure p_{1t}. If this expansion is carried out with an efficiency η_j, then an equation analogous to equation (3) gives:

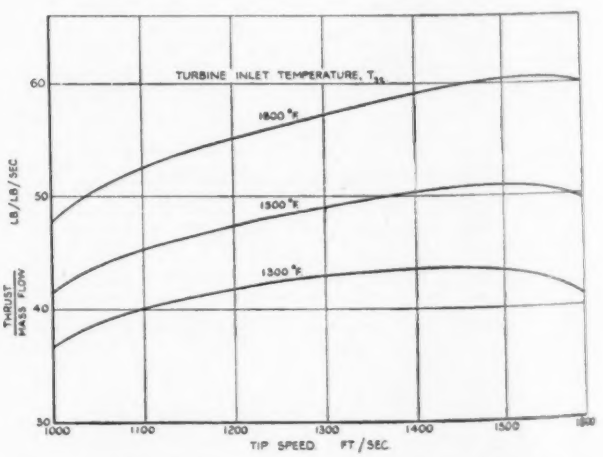
$$\frac{p_{1t}}{p_{4t}} = \left(1 - \frac{1}{\eta_j}\ \frac{T_{4t} - T_5}{T_{4t}}\right)^{\frac{\gamma}{\gamma - 1}} \tag{6}$$

Here T_5 is the actual temperature of the gases after expansion to atmospheric pressure is complete. The whole of the temperature drop is converted into kinetic energy and:

$$V_5 = \sqrt{2g\ J\ C_p\ (T_{4t} - T_5)} \tag{7}$$

where J is the mechanical equivalent of heat and C_p is the mean specific heat at constant pressure over the expansion.

Consequently, from equations (6) and (7), V_5 is expressible in terms of p_{4t}, p_{1t} and T_{4t}. Using the relations for p_{4t} and T_{4t} given in equations (3) and (4) the thrust per unit rate of mass flow is:



■ Fig. 10 – Thrust per unit rate of airflow through jet turbine

$$\sqrt{\frac{2\eta_j J C_p}{g} T_{3t}\left(1-\frac{0.9\,(T_{2t}-T_{1t})}{T_{3t}}\right)\left(1-\frac{(p_{1t}/p_{3t})^{\frac{\gamma-1}{\gamma}}}{1-\frac{0.9\,(T_{2t}-T_{1t})}{\eta_t\,T_{3t}}}\right)} \quad (8)$$

This relation is graphically represented in Fig. 10 as a plot of thrust per unit mass flow (lb per lb per sec) against compressor tip speed for various values of T_{3t}

The values taken for the constants are:

$\gamma = 4/3$ $\quad \eta_t = 0.87$

$\eta_j = 0.95$ $\quad J = 778$

$C_p = 0.27$

The compressor slip factor is taken as 0.91 and the values of pressure ratio have been taken from Fig. 4. The turbine inlet pressure p_{3t} has been calculated assuming that the loss of total pressure in the combustion chamber is 3% of the absolute pressure p_{2t} at inlet.

The thrust per square foot of exhaust annulus area may now be obtained by multiplying the thrust per unit mass flow [equation (8)] by the mass flow per square foot of annulus area derived from equations (1), (2), (3), and (4).

In Fig. 11 the thrust per square foot of turbine exhaust annulus is plotted against impeller tip speed for various turbine exhaust Mach numbers M_4 and turbine inlet temperatures T_{3t}.

Another important factor, which may be expressed as a function of the exhaust annulus area, is the centrifugal stress in the turbine buckets. If the buckets are unshrouded, unflared, and parallel sided, that is, of constant cross-sectional area from root to tip and with the leading edge equal in height to the trailing edge, the stress in psi at the root due to centrifugal force is:

$$1.9\,A\left(\frac{N}{100}\right)^2 \quad (9)$$

where N is the rpm of the turbine rotor and the density of the bucket material is taken as 0.29 lb per cu in. This relation is derived in standard textbooks on the steam turbine.

Relating the impeller tip speed U in the compressor to the rotor rpm N by means of the impeller tip diameter D, the stress in psi per sq ft of exhaust annulus area becomes:

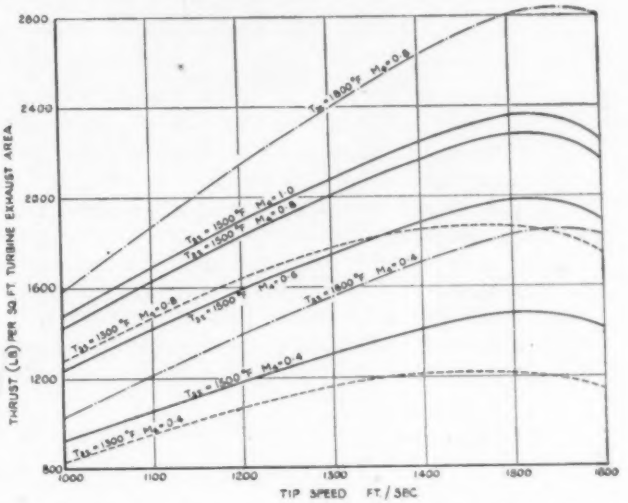
$$\frac{1.9}{D^2}\left(\frac{60\,U}{100\,\pi}\right)^2 \quad (10)$$

The curves for thrust per square foot of exhaust annulus (Fig. 11) may be combined with the relation for stress given in equation (10) to give a relation between thrust and stress. This relationship is shown in Fig. 12 as a plot of thrust per (stress x D^2) versus impeller tip speed for various exhaust Mach numbers M_4 and turbine inlet temperatures T_{3t}.

For given values of M_4 and T_{3t} the ratio between thrust and stress is proportional to the square of the impeller tip diameter. For a given stress and impeller tip diameter the thrust increases with exhaust Mach number and temperature. Increase in Mach number above 0.8 is difficult because of the great increase in aerodynamic losses in the exhaust ducting and even in the turbine buckets. Below a Mach number of 0.6 the thrust drops rapidly and there may be difficulty in obtaining the requisite heat drop in the turbine without appreciable residual swirl in the gases leaving the buckets. The choice of turbine inlet temperature T_{3t} depends on the creep properties of the bucket material. In general, due to their relative motion the buckets run at a temperature some 220 F to 280 F lower than the turbine inlet temperature. For a given thrust a relation between stress and temperature in the buckets may be obtained from Fig. 12. When this is compared with the safe working stress for the bucket material at various temperatures, deduced from creep tests, it is often found that a higher factor of safety can be obtained by reducing the temperature and raising the stress rather than by the reverse procedure.

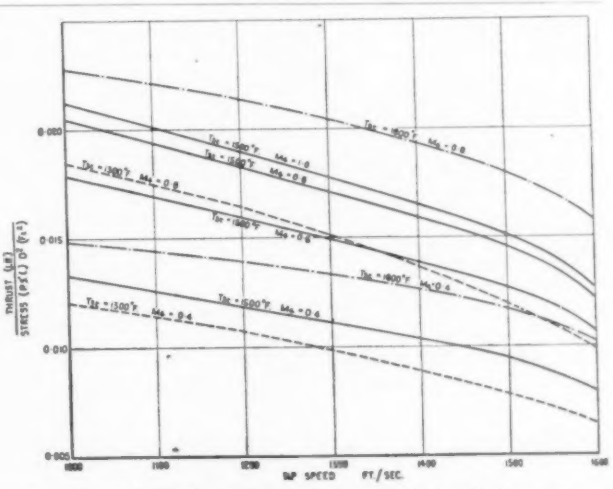
The stresses so far discussed have been those in a parallel-sided bucket. By decreasing the cross-sectional area of the bucket from root to tip the centrifugal stresses may be appreciably reduced. This is generally done by an approximately linear reduction of cross-sectional area with bucket height. The reduction in stress which may be achieved is shown in Fig. 13. Better results may be obtained by reducing the area parabolically with bucket height, although this increases manufacturing problems.

Values for existing jet turbines are shown in Fig. 13 and



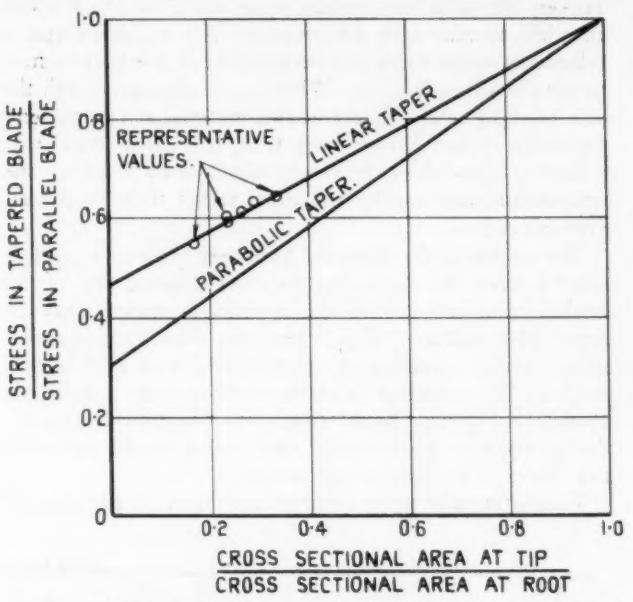
■ **Fig. 11 – Thrust per unit cross-sectional area of turbine exhaust annulus**

T_{3t} = Turbine inlet temperature
M_4 = Mach number in exhaust annulus



■ **Fig. 12 – Relation between thrust and centrifugal stress at root section of untapered turbine buckets**

T_{3t} = Turbine inlet temperature
M_4 = Mach number in exhaust annulus



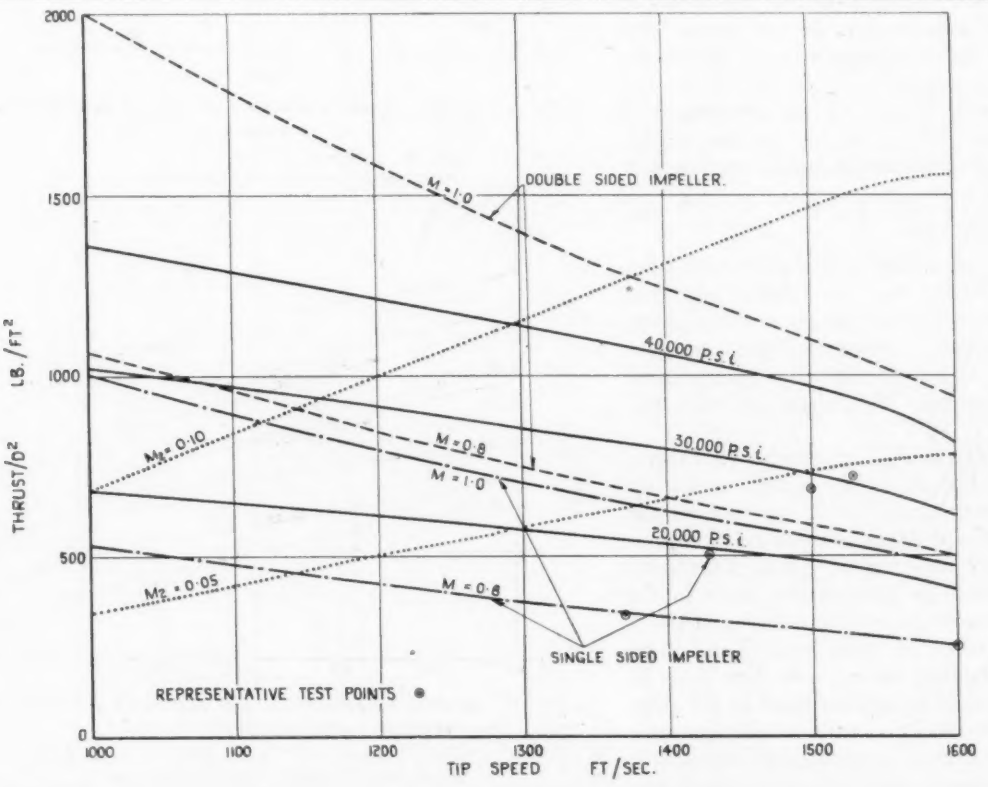
■ Fig. 13 – Effect of tapering reduction in area on centrifugal stresses in turbine bucket

indicate that a reduction of stress to 0.6 of the value for the parallel-sided bucket is possible.

Using a value of bucket stress equal to 0.6 of the value for the parallel-sided bucket, some curves relating the value of thrust per D^2 to impeller tip speed and bucket stress are given in Fig. 14. In the following discussion, thrust per D^2 will be referred to as the "thrust factor." It is the thrust obtainable from an engine with an impeller one foot in diameter. To simplify the diagram it has been drawn for an exhaust Mach number of 0.8 and a turbine inlet temperature of 1500 F only. Fig. 14 shows that, for a given stress in the buckets, higher thrusts are obtained at lower tip speeds. This is because, as the tip speed is reduced, longer buckets may be fitted to the turbine for the same stresses and consequently the mass flow may be increased.

As the mass flow is increased the conditions in the inlet to the compressor may become a limiting factor. The curves of Figs. 6 and 10 may be combined to give a curve of thrust factor against tip speed for various values of the Mach number M at the tip of the inlet vanes on the impeller. These are shown in Fig. 14 for values of $M = 0.8$ and 1.0. Two sets of curves have been drawn. One (broken lines) for a double-entry impeller of the type shown in Fig. 1, and the other (chain dotted lines) for an engine with a single-entry impeller. In both instances, if for reasons of efficiency the Mach number M is limited to a certain value, more thrust is obtained at the lower tip speeds.

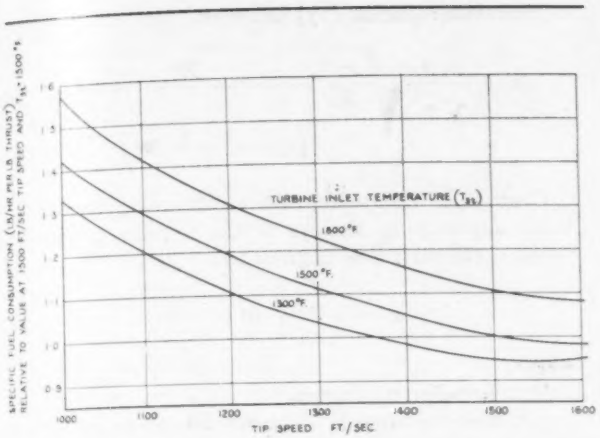
Fig. 14 enables a comparison between engines with single- and double-entry impellers to be made. If the designer chooses a double-entry impeller, stresses in the turbine buckets are likely to be the limiting factor. If a single-entry impeller is chosen, the limitation is likely to occur at the entry to the impeller. The choice is then between an aerodynamic and a stress limitation. At the moment the development of materials appears to be ahead of our knowledge of aerodynamics, for the highest values of the thrust factor are obtained from jet engines with double-sided impellers. This does not, however, complete the comparison. The use of double-sided impellers, because it makes the air intake ducting more complicated, transfers the aerodynamic problems to the aircraft designer. In the engine with the single-sided impeller pro-



■ Fig. 14 – Principal aerodynamic and stress limitations to thrust obtainable from jet turbine

M = Impeller Mach number at impeller tip

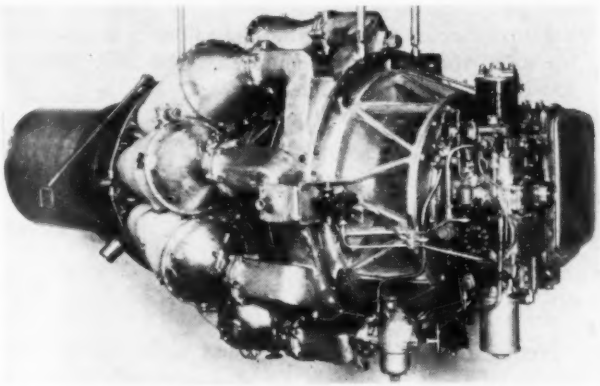
M_2 = Mach number in combustion chamber



■ Fig. 15 – Effect of impeller tip speed on specific fuel consumption in jet turbine

vision does not have to be made for flow around the compressor casing and consequently intake design is simplified and the ratio of the overall diameter of the engine cowling to the impeller diameter may be reduced. The comparison, therefore, should take into account the problems of installing the engine in the aircraft which are becoming more serious as aircraft speeds increase.

Another limitation to the thrust factor obtainable from the engine may arise due to the flow conditions in the combustion chamber. It is possible to design a compressor casing whose outside diameter does not exceed about 1.75 times the diameter of the impeller. It is desirable to keep the combustion chambers within this diameter in order to avoid an increase in frontal area. If, for example, nine combustion chambers are used, their cross-sectional area will be (from Fig. 8) about 46% of the total frontal area. The combustion-chamber Mach number M_2 may be related to the mass flow flowing through this area, hence to the mass flow for a given tip diameter and finally to the thrust factor. Because M_2 is generally low the thrust factor is proportional to it. In Fig. 14, using the above assumptions, curves of the thrust factor for values of combustion-chamber Mach number of 0.05 and 0.10 have been drawn. Attention is drawn to the original assumption of 3% combustion-chamber pressure loss used in making the calculations for Fig. 14. It is difficult to prevent this loss from increasing if values of M_2 above 0.05 are used. The use of higher values and correspondingly higher losses will necessitate some revision of the curves of Fig. 14 in the direction of lower thrusts.



■ Fig. 16 – Rolls-Royce Derwent V (design commenced on Jan. 1, 1945; captured world speed record on Nov. 7, 1945)

Consequently, the conclusion that it is preferable to use low tip speeds, both from the point of view of Mach number at the impeller inlet and stresses in the turbine blading, must be regarded cautiously, because it may lead to an increase in frontal area of the engine in order to maintain low combustion pressure losses. If the combustion-chamber dimensions are to be kept within the frontal area of the compressor high losses may result with a lowering of thrust and fuel economy.

The specific fuel consumption of the engine also increases as the tip speed is reduced. The curves of Fig. 15, based on the same assumptions as those used in Fig. 14, show how the specific fuel consumption (lb per hr per lb thrust) is affected by choice of impeller tip speed and turbine inlet temperature. The curves in Fig. 15 are drawn to show values of the specific consumption relative to the value at 1500 fps and 1500 F. Below tip speeds of the order of 1400 fps the fuel consumption rises appreciably.

Some values of the thrust factor of current engines are shown in Fig. 14. Because they are not necessarily operating with the turbine temperature and the exhaust Mach number chosen in the preparation of Fig. 14, the test bench values are not strictly comparable with the curves. However, they show the trends of current design practice. The value at which Whittle's W1 engine[1] first flew was 340 (psf units). Present-day values for engines with double-entry impellers are over double. Engines with single-entry impellers give values over 500. The de Havilland Goblin III has a thrust factor of 525. The Rolls-Royce Derwent V engines (Fig. 16) used to take the world's speed record were rated at a value just below 1000. It will be seen from the points on Fig. 14 that designers have chosen to use tip speeds of about 1500 fps as representing the best compromise between size, weight, and fuel consumption. Lower tip speeds might be used for special engines, where high thrust is more important than fuel economy.

■ Conclusion and Acknowledgments

By concentrating attention on the simplest type of aircraft gas turbine and by selecting certain aspects of its design for analysis, some illustration has been given of the nature of its problems. Several problems have, perforce, gone unmentioned. Those associated with the vibration of buckets and vanes and with the critical speeds of rotors rank as the most important. Thermal expansion problems are, perhaps, next. They all yield fitfully to analysis. Bearings, seals, controls, and lubrication form a substantial tail which often wags the dog.

But this paper has emphasized that the major conflict in the design of the gas turbine is, very briefly, the conflict between Mach number and stress. Research is needed in two main directions. In metallurgy there is much yet to be achieved in developing materials to withstand creep and fatigue, oxidation and erosion at ever higher stresses and temperatures. In aerodynamics compressible flow is the greatest field of research. The control of gases moving at high speeds must be perfected. The aerodynamics of the combustion process must be better understood and the knowledge applied.

Workers on these problems have made possible the present achievements of the aircraft gas turbine. It is upon

the success of their continued work that future progress depends.

The author acknowledges with thanks the help of J. M. Shaw of the Royal Aircraft Establishment who made the calculations and prepared the diagrams. Fig. 1, the general arrangement of the Derwent I, and Fig. 16, the Derwent V which took the world speed record recently at 606 mph, were made available by Rolls-Royce, Ltd. The de Havilland Aircraft Co., Ltd., supplied the data on the Goblin, which has seen many hours flying in America. The courtesy of these organizations is gratefully acknowledged.

APPENDIX

The mass flow through a duct of cross-sectional area A is given by:

$$W = \rho A\ V \tag{11}$$

where ρ is the density of the gas and V the velocity. The velocity of sound may be expressed as:

$$\text{Velocity of sound} = \sqrt{g\gamma RT} \tag{12}$$

where T is the absolute temperature, R the gas constant, g the acceleration due to gravity, and γ the ratio of the specific heats. Whence the Mach number:

$$M = \frac{V}{\sqrt{g\gamma RT}} \tag{13}$$

Also the density ρ is related to the absolute pressure p and T by the gas equation:

$$\frac{p}{\rho} = RT \tag{14}$$

Whence equation (11) may be written:

$$W = p\ M \sqrt{\frac{g\gamma}{RT}}\ A \tag{15}$$

The values of p and T given are the values which would be measured by an observer moving with the stream. The stagnation values p_t and T_t are related to these by the expression:

$$\frac{p}{\rho_t} = \left(\frac{T}{T_t}\right)^{\frac{\gamma}{\gamma-1}} \tag{16}$$

Substituting for p in equation (15):

$$W = \frac{p_t A}{\sqrt{T_t}}\ M \sqrt{\frac{g\gamma}{R}\left(\frac{T}{T_t}\right)^{\frac{\gamma+1}{\gamma-1}}} \tag{17}$$

Now from the principle of conservation of energy:

$$T_t = T + \frac{V^2}{2gJC_p} \tag{18}$$

where J represents the mechanical equivalent of heat and C_p the specific heat at constant pressure, then:

$$\frac{T_t}{T} = 1 + \frac{M^2 g\gamma\ RT}{2g\ J\ C_p\ T} = 1 + \frac{\gamma - 1}{2} M^2 \tag{19}$$

Therefore equation (17) becomes:

$$W = \frac{p_t A}{\sqrt{T_t}} \sqrt{\frac{g\gamma}{R} \frac{M^2}{\left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{\gamma+1}{\gamma-1}}}} \tag{20}$$

Centrifugal Compressor Inlet – Assuming that the velocity approaching the eye of the impeller is axial and uniform the mass flow is given by:

$$W = \rho\ V_a \frac{\pi}{4} D^2 (e^2 - h^2) \tag{21}$$

where:

D = Impeller tip diameter

e = Ratio of outside diameter of eye to impeller tip diameter

h = Ratio of inside diameter of eye (hub diameter) to impeller tip diameter

The subscript a refers to the axial velocity approaching the impeller eye.

From a consideration of the velocity triangles the maximum velocity and Mach number relative to the rotating impeller vanes (assuming they are of negligible thickness) occur at the outer diameter of the eye where the relative velocity attains the value:

$$V_r = \sqrt{V_a^2 + e^2u^2} \tag{22}$$

where u is the velocity of the impeller tip (diameter D) and the Mach number:

$$M = \frac{V_r}{\sqrt{g\gamma RT}} = \sqrt{M_a^2 + \frac{e^2u^2}{g\gamma RT_t} \frac{T_t}{T}} \tag{23}$$

Now substituting a relation of the form of equation (19) in equation (23) and rearranging the terms to solve for M_a:

$$M_a^2 = \frac{M^2 - \frac{e^2u^2}{g\gamma RT_t}}{1 + \frac{\gamma - 1}{2} \frac{e^2u^2}{g\gamma RT_t}} \tag{24}$$

Rewriting equation (21) in the form of equation (20):

$$\frac{W\sqrt{T_t}}{p_t \frac{\pi D^2 (e^2 - h^2)}{4}} = \sqrt{\frac{g\gamma}{R} \frac{M_a^2}{\left(1 + \frac{\gamma - 1}{2} M_a^2\right)^{\frac{\gamma+1}{\gamma-1}}}} \tag{25}$$

For convenience in plotting, the values of mass flow and tip speed may be corrected to standard inlet conditions, 520 R and 14.7 psi, absolute.

Then:

$$\frac{W_{corr}\sqrt{520}}{14.7 \times 144} = \frac{W\sqrt{T_t}}{p_t} \tag{26}$$

and:

$$\frac{U_{corr}}{\sqrt{520}} = \frac{u}{\sqrt{T_t}} \tag{27}$$

Noting that the velocity of sound under standard conditions is:

$$a = \sqrt{g\gamma R\ 520} \tag{28}$$

equation (24) becomes:

$$M_a^2 = \frac{M^2 - \frac{e^2u^2}{a^2}}{1 + \frac{\gamma - 1}{2}\frac{e^2U^2}{a^2}} \qquad (29)$$

on inserting this in equation (25) and rearranging the terms.

$$\frac{W_{corr}}{D^2}\frac{e^2}{(e^2 - h^2)}\frac{U^2}{a^2} = \frac{4}{\pi}\frac{14.7 \times 144}{\sqrt{520}}\frac{e^2U^2}{a^2}$$

$$\sqrt{\frac{g\gamma}{R}\frac{\left(M^2 - \frac{e^2U^2}{a^2}\right)\left(1 + \frac{\gamma - 1}{2}\frac{e^2U^2}{a^2}\right)^{\frac{2}{\gamma-1}}}{\left(1 + \frac{\gamma - 1}{2}M^2\right)^{\frac{\gamma+1}{\gamma-1}}}} \qquad (30)$$

For convenience of presentation the relationship is shown in Fig. 5 in the form of a plot of

$$\frac{1}{(1 - h^2/e^2)}\frac{W_{corr}}{D^2}\left(\frac{U}{1400}\right)^2$$

against $\frac{eU}{1400}$ for various values of M, the Mach number relative to the rotating inlet vanes at the tip.

DISCUSSION

Points Out That Analysis Is Applicable to Single Design

—KENNETH CAMPBELL
Wright Aeronautical Corp.

DR. HAWTHORNE states early in the paper that he is basing it on a quite predetermined choice of design; and we infer that in this design the achievement of the high power-weight ratio mentioned was paramount among all other considerations. Restriction of the paper to this design has necessitated foregoing some qualifying factors of great importance to the student of designs as a whole and we mention one or two of these.

Figs. 5 and 6 are helpful in selecting the optimum inlet/OD ratio of an impeller with respect to air handling capacity; but reliance on the larger values of abscissa of Fig. 5 is accompanied by serious reduction in efficiency because of an inlet above limiting proportions in this respect. Fig. 6 gives maximum air handling capacities based on peak values of the curves of Fig. 5; but the entire upper half of Fig. 6 is in the region of those heavy efficiency penalties. Curves of limiting inlet/OD values might be plotted across these curves to warn the designer. By designing somewhat to the left of the peaks of Fig. 5, reflecting more conservative inlet proportions, a new Fig. 6 could then be plotted. This would be a compromise in maximum handling capacity between the values shown and the severe limitations just pointed out.

Some emphasis has been placed on the merits of the impeller having double inlet, running at a very high tip speed to attain the pressure ratio desired from a single stage. From a strictly power-weight ratio standpoint there is little argument. From an efficiency standpoint, we wonder whether the value of 75% quoted, excellent for 1500 fps tip speed, is inclusive of the induction losses to the rear impeller. We do not refer to the aircraft designer's passages mentioned but from just outside the machine radially inward past the longitudinal ducting and other basic structure.

On capacity, at tip speeds of 1500 fps and higher, inlet diameter must be reduced below maximum proportions governing efficiency from other than Mach number considerations. This is so as not to exceed even the high Mach number of 9, let alone more conservative values. Purely to illustrate the difficulties of considering only one design, if two single-inlet impellers of this OD are run in series to achieve this same pressure ratio, their tip speed and inlet blade speed are so greatly reduced that the maximum inlet normally allowable can be used. The capacity equals that of the double-inlet impeller at the same inlet Mach number.

We like the author's suggestion of a combustion-chamber Mach number parameter but we must be cautious in defining it. Should some other shape of chamber be considered, for example, whereby the largest cross-section occurred much later in the process of adding heat and air, a cold air Mach number at this point might mean substantially higher cold air Mach number at the beginning of the process, and limiting values found for the proposed parameter might differ considerably from Dr. Hawthorne's values.

The analysis in the paper results in an illuminating presentation of the thrust-stress ratio relations, but the surveyor of the general problem should recognize its qualitative nature. For example, assume some other compressor design giving the same pressure ratio at substantially lower tip speeds, and perhaps with substantially higher capacity. A somewhat larger turbine wheel (to illustrate, one with buckets of the same height and tip speed) would provide a larger annulus, capacity, and thrust for the same stress. The combustion chamber would perhaps be less space limited for reducing combustion dynamic loss, first recognition of which we believe is one of the author's many contributions.

Such a step would involve a reevaluation of the reduced thrust-weight ratio and, equally important, the substantially increased efficiency of components, or improved fuel consumption. Here the problem becomes complex and controversial, but we have introduced these comments merely to accentuate the difficulties in drawing too tight a conclusion from study of a specific design. Let us acknowledge, though, that inclusion of all Dr. Hawthorne's thoughts on this subject would have constituted a paper of prohibitive length.

AUTHOR'S CLOSURE TO DISCUSSION

IN his very welcome comments Mr. Campbell has raised some interesting points. His discussion on the problem of compressor intake design, backed as it is with his extensive experience in the problems of compressor development, usefully confirms my selection of this as one of the aerodynamic key points in a jet turbine. He has rightly pointed out that the addition of an extra stage very much simplifies the designer's problems; nevertheless the development of efficient impellers with a large capacity is likely to be actively pursued if only because of the general tendency to call for larger and larger powers from any given aircraft engine during its development period. The figure of 75% at a tip speed of 1500 fps is inclusive of the induction losses described by Mr. Campbell but does not allow for any heating of the air flowing into the rear intake which results from its contact with hot casings.

Mr. Campbell has pointed out that the results of the analysis contained in the paper are only applicable to one design. The methods used in the analysis are, however, easily applied to any design. Compare, for instance, the design described in the text with one comprising a two-stage compressor of the same pressure ratio and tip diameter. The tip speed will be reduced $\sqrt{2}$ times and the mass flow will be doubled. Either the combustion-chamber Mach number will be doubled and the combustion pressure losses quadrupled or the overall diameter of the combustion chambers will have to be increased $\sqrt{2}$ times to maintain the same losses as those in the single-stage unit. Because of the lower rotational speed, the rim diameter of the turbine will have to be increased $\sqrt{2}$ times to maintain the work output. The stress in the turbine blades remains the same because, although the rpm is reduced, the turbine exhaust area is increased to pass the higher mass flow (compare equation (9) in text of paper). For these reasons there are only marginal differences in efficiency and power-weight ratio between a single- and two-stage compressor unit working *at the same pressure ratio*. If the pressure ratio in the two-stage unit is increased above that of the single-stage unit some gains are to be expected. Further extension of the analysis to units using the axial-flow compressor suggests that at present the combustion chamber presents the largest cross-sectional area and determines the overall diameter of the engine. Turbine weights are likely to be higher in axial-compressor engines, although the turbine diameter may be decreased by the use of double instead of single stages.

The Philosophy of C

THE basic philosophy of cooperative research is discussed by Mr. Veal, as well as the means developed by experience through which these functions can be exercised most expeditiously. The research examined here is that accomplished through voluntary association among industrial organizations, not the government initiated and supervised activity of war's emergency.

After Mr. Veal received the Horning Memorial Award for 1944, he delivered this lecture on cooperative research.

by C. B. VEAL

Coordinating Research Council, Inc.

IN accepting the Horning Memorial Medal Award, I disclaim individual credit. The award belongs to those who have worked actively in CRC and supported it financially. This year's award does not signalize specific achievement in engine design or petroleum chemistry leading to better mutual fuel-engine performance. It is a tribute to the accomplishments of the cooperative research of the automotive and petroleum industries which *he,* in whose memory this award is given, was instrumental in *initiating* and which *he* so ably supported.

For this reason, *cooperative research* is appropriately the theme of this lecture. Its purpose is to present certain considerations concerning the basic functions of cooperative research, and the means, developed by experience, through which these functions may be exercised most expeditiously. The research here discussed is that accomplished through the voluntary association among industrial organizations, not the government initiated and supervised activity of war's emergency. Frequent reference will be made to the joint research of the automotive and petroleum industries. This is understandable in view of my 20 years' association with this project, in its original form as the Cooperative Fuel Research Committee, celebrating its twenty-fifth anniversary this year, and in its expanded and independent functioning, since May 1, 1942, as the Coordinating Research Council, Inc.

Appropriate to the times, and indeed imperative, is such a reappraisal of aims and determination of means. The war has driven us hard and unremittingly toward a single, specified end. It has called out our utmost capabilities, but it has suppressed choice of gait and goal. Having passed the finishing line victoriously, we are again free to follow the paths of our choice. In this period of rededication, an alert understanding of objectives is vital. The taking of a wrong fork in the road, though it, in the beginning, may diverge only slightly from the true path, eventually may lead far from the designed destination. A general confusion and disturbance results from the relaxation of war restrictions and from postwar readjustments. To sift out the *fixed fundamentals by which to hold* calls for calm and concentrated reflection. The lessons learned in World War I led to the founding of CFR, initially through the inspiration of Harry L. Horning and Henry L. Doherty.

Four organizations sponsored the war-born infant, the American Petroleum Institute and the National Automobile Chamber of Commerce, which furnished the funds, the Society of Automotive Engineers, which provided the secretariat, and the National Bureau of Standards, representing the various interested Federal agencies and making available its unique laboratory facilities and personnel. The interest of the Bureau of Standards in the development of the CFR enterprise as a closely confederating instrumentality with Government agencies was manifested in the granting to Dr. H. C. Dickinson, chief of its Division of Heat and Power, leave of absence to serve as the first Research Manager of the SAE, and, in this capacity, as the first secretary of the CFR Committee.

The depression years following 1929 brought about the transition from the early method of executing projects, under the direction of CFR, at the Bureau of Standards, with funds provided by industry through the NACC and the API, to the present method of having members of Groups planning the research carry it out in their own laboratories.

The third high point in the development of the cooperative research was the creation of the Coordinating Research Council, comprising the existing CFR Committee and the two newly created technical committees, the Coordinating Lubricants Research and the Coordinating Equipment Research Committees.

The fourth stage is now at hand – what will the aftermath of World War II do to CRC?

■ Sphere of Cooperative Research

A criticism sometimes leveled against cooperative research is lack of achievement in the outposts of *science*

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 10, 1946.]

COOPERATIVE RESEARCH

THE AUTHOR: C. B. VEAL (M '12), who was SAE research manager and secretary of the CFR Committee for 16 years, has been manager and secretary of the Coordinating Research Council since it was organized in 1944. During the early part of his career he had an extensive consulting practice, and he was also the head of Purdue University's machine design department for several years before World War I. Mr. Veal, who is a prolific writer of technical articles, has had many of them published in the *SAE Journal*.

and *design*. Such criticism reveals a misunderstanding of the true sphere of cooperative research. The outposts of scientific research are best occupied by experimenters of singular brilliance and organizations with specialized equipment. New designs originate from men of creative genius, the inventors, and are developed by commercial organizations. Cooperative research, involving the united efforts of many persons, must not aspire to the role of the pioneer. Since it represents many different industrial organizations, it must not infringe upon the design and development prerogatives of the individual industrial concern. *In the broad general or collective application of the principle and the utilization of the design lies the function of cooperative research.* It must bring such knowledge into the *forefront of technical consciousness* and, spreading it among greater numbers, give it the familiarity which leads to understanding *assimilation, appreciation,* and *general utilization*. It follows from these considerations that the subject matter and findings of cooperative research will seldom be such as to catch the public eye. They will lack both startling novelty and direct interest to the layman. This *does not mean that the public receives no benefit* from cooperative research; *far from it.* Cooperative research serves the industry or industries which support it, and, through industry, the general public. In fact, the extent to which any newly proposed project meets this requirement is a major factor in determining whether or not it is to be undertaken as cooperative research.

As handmaiden to the gods of *science and design,* cooperative research, while humble, significantly accelerates the onward march of technical progress. This advance has always been halted by that *weary wait* between the discovery of a new bit of knowledge and its general application. A contemporary scientist when asked the difference between pure science and industrial research replied: *"twenty years."* The ancient Greeks knew that the earth was round, and that it moved, yet in the sixteenth century Galileo was persecuted for presenting these facts, unpalatable to a wrong-headed world, and only today is knowledge of the earth's conformation being fully utilized in the laying out of air transport routes. The principles involved in the release of atomic energy have long been scientific knowledge, but only the impetus of war and an organized research of vastly unprecedented proportions incorporated them in a form helpful to *G. I. Joe.* In our own field, Rudolf Diesel for many years was refused a hearing for his engine and met his tragic and mysterious death before the principle of compression-ignition internal-combustion engine operation was accepted. Cooperative research, defined as *purposeful organized investigation* in the field of the general application of principles embracing the efforts of all interested and qualified sections of industry, will help bridge the gap between the *known* and the *used*.

A frequently asked question is, what criterion should be applied to determine whether or not a project should be undertaken. The answer to this is *simple, certain,* and *practical,* although sometimes shocking to the idealist questioner. A project should be undertaken only when *industry is willing to pay for it.*

That needed knowledge is lacking on a particular topic is no justification, in itself, for carrying out cooperative research. The knowledge required may be of limited applicability, or it may be ahead of industry's thinking; in either case it will not be used or useful even if developed. If industry is willing to pay for a cooperative project, that project is born with three inherent elements of successful research: *the knowledge it is planned to provide is needed, that need is widely felt,* and the *results obtained will be immediately accepted and used*. Occasionally, a worthy project, because of bad timing or inadequate presentation, is rejected by industry, but such projects profit by delay and eventually survive. It is always true that any line of investigation that needs to be *sold* to industry *needs itself to be investigated;* for no cooperative research *so foredoomed* to *sterility* should be undertaken.

A topic becomes ripe for cooperative research when a certain level of common knowledge concerning it has been reached. When possible participants know little or nothing about a topic, they are unwilling to join a project because they have too little to contribute, and because they do not wish to expose to others, particularly their competitors, the extent of their ignorance. A group meeting held under such conditions results either in a *silence*

equalled only by a *melancholy Quaker Meeting*, or in endless *twaddle* matched in *worthlessness* by the traditional vaporings of an Old Ladies Knitting Society. When each prospective participant, in his laboratory, has pursued a certain line of investigation with some success, but has gone as far as he can at the time, he may feel that he can both contribute to and gain from a meeting with other workers on the same subject. In a discussion held at this state of affairs, the various viewpoints will be presented fully and a level of common knowledge established to serve as the basis for *future cooperative progress*. A selection may be made of the approach which appears to offer the most promise, and a cooperative project is under way.

Historically, the CFR and the CLR exemplify this analysis. Cooperative research was possible on motor gasoline before it could be undertaken for diesel fuel or aviation gasoline. Motor crankcase oil could not be handled until later. It took the exigencies of war to produce the first truly effective cooperative endeavor on motor oils and gear oils, and even now little is being accomplished on aviation-engine oils.

■ Tools of Cooperative Research

One of the handiest tools of cooperative research is the statistical method. Facts can be gathered in significant volume to prove or disprove a thesis, while the question is still of urgent interest. The first project undertaken by the Cooperative Fuel Research involved application of the statistical method. This project was to determine what volatility limits should be set on fuels to produce the greatest number of car miles per barrel of crude oil under the then current operating conditions. Service mileages were accumulated by owner-driven car operation reported to nine cooperating companies, and by test runs supervised by the National Bureau of Standards with equipment and men loaned by cooperating companies. The collection of data by the various CFR detonation Exchange Groups is an excellent example of the application of the statistical method. Four analyses of these ratings, made in 1936, 1939, 1942, and 1944, have, for motor fuels alone, included *16,968 tests on 468 fuels*. Paging through the current CRC annual reports yields numerous examples of the speedy accumulation of large amounts of data needed to solve statistically some knotty problem of military equipment operation.

A second instrument of cooperative research is the attainment by technical men of a common understanding on the meaning of general terms in their specific field. In its simplest form, this understanding is manifested in the establishment of a generally acceptable vocabulary. Engineers are prone to be more interested in materials and observed phenomena than in words. Absorbed in experimental interests, they christen a device or a condition with what seems an apt descriptive term. Thus, in fuel studies, we have "fish hook" curves, "twiddlers," detonation "puffs," "regurgitators," "pings," "gunk," "C.S.," boost pressures, 1000-ft detonation intensity. Various pieces of apparatus have become known as "squirrel cages," "egg beaters," "mix masters," "nervous counterweights," and other expressions such as "trace knock," "burned piston," and "pickup" have acquired their own definite meanings. In large research organizations, over a period of years, several different terms may be evolved having essentially the same definition. This can lead to duplication of effort within associated laboratories, and to confusion and lack of ability to interchange ideas among men of different organizations. Where men who have common interests in a highly specialized field meet frequently for discussion, as in cooperative research projects, a common vocabulary is developed. A prime example of this is given in the term "mixture response curve," which is markedly different from the term "mixture control curve." These terms, while in nowise interchangeable in technical meaning, are often by common usage interchanged by individuals who have not had the benefit of cooperative work. The importance of this tool of expression and communication was emphasized by C. F. Kettering in a recent discussion. He said: "We are getting into the Tower of Babel as far as technological lingo is concerned, and we need to get down into common words. Some time ago an effort was made to prepare a dictionary of scientific terms. As I recall, we got as far as the word 'atom.' We didn't get any further because we couldn't reach any agreement. Then they finished it up by calling it 'a glossary of the probable meaning of these words.'"

When the item to be defined is a property or a characteristic of a material, or a complexity of operating conditions, arrival at a common understanding is a more extensive and burdensome process than the mere formulation of a vocabulary. Words applied to such properties or conditions represent definite conceptions only if they comprise a commonly accepted understanding of the manner of their determination. This leads to a review of the factors entering into such a determination – temperature, pressure, time – and the selection of the critical points in such factors that have general practical significance. The final step is to draw up a test procedure incorporating these factors in such form that its several provisions may be understood and applied readily. These procedures must yield results that are both significant and reproducible. To develop this quality of reproducibility is *preeminently* a function of cooperative research with its command of the experimental and statistical methods. *Reproducibility* is the *first objective* in the development of a procedure. Its significance in correlation with service performance is secondary, for a test, *if reproducible, is always useful*, although experience may demonstrate that its best application is other than that originally expected. A further possible step is simplification of the procedure, the devising of short-cut operations or apparatus to yield the same results, or of instruments to measure them.

That such procedures must not be confused with standards or with purchase specification criteria cannot be too often emphasized. The establishment of a standard involves its acceptability from a commercial standpoint; a purchase specification must be controlled by specific economic conditions involving the influences of supply and demand. Procedures evolved by cooperative research are concerned with the definition of basic properties; they are the *instruments of research, not of commerce*.

In this connection, it should be noted that the CRC is devoted *exclusively to research*, hence rigidly limits its working memberships to technical men, preferably those engaged in research and development. Since commercial representatives are not welcome in its groups, it is basically entirely unfitted to prepare specifications and standards, with their inherent commercial implications.

The type of procedure discussed above is indeed a *"reproducible experiment"* as the term was used by Thomas Midgely in the last paper he presented before his untimely death. It may also claim the *importance* attributed by Dr. Midgely to the "reproducible experiment," when he said: "To my mind the basis of the scientific process is the reproducible experiment.

"Facts are still, and probably always will be, determined by vote, quite as the College of Cardinals determined the number of feathers in the Archangels' wings just a few centuries ago . . . Indeed the same process is in use today when groups of scientists gather to hear discussions of controversial subjects. There are, however, two points of basic difference. Whereas the number of feathers was decided by majority vote, in science we require a practically unanimous vote for establishing a fact. The second point is the type of evidence required by the voter to influence his decision. Revelations, dreams, supernatural authority are now out, and even logic is of secondary importance to the *reproducible experiment.*"

Let me cite one example among the cooperative achievements of the petroleum and automotive industries in establishing a commonly accepted vocabulary. This is the term "octane number" defining the knocking characteristics of a fuel or an engine. The term originated in an individual industrial laboratory. It acquired significance through the ascertainment of and agreement on the conditions under which it was to be determined, a continuing project of the CFR. It became more generally acceptable as the CFR Cooperative Exchange tests accumulated data in statistical proportions on its reproducibility over a period of a dozen years, and it has worked its way into the vocabulary of the filling station and the motoring public. Today, only two arbitrary systems of measurement are universally accepted; one measures time, the other, octane number. No greater tribute can be paid to CFR research results than that today the manufacturing departments of the oil industry throughout the world operate their refineries without question in blind adherence to the knock-rating procedures provided by the Committee's cooperative research.

Forty-seven CRC *test procedures* are in active, current use; 44 of these are presented in the 1946 CRC Handbook.

A third tool of cooperative research is education. For the younger technical men, Group discussions constitute a postgraduate course, bridging the gap between the theories of school and the practices of industry. Cooperative enterprises, presenting the viewpoints and methods of many different laboratories, are pointedly instructive and promote the flexibility of mind that is a prerequisite for mental development and scientific progress.

Supplementary educational agencies for the present generation of young technical men are of overwhelming importance. Young men who ordinarily would have been in technical universities have been learning *grimmer lessons* in *tougher schools*. Even those whose duties have been of an engineering nature or who have taken the wartime accelerated engineering courses are likely to need special reorientation to the conditions and requirements of peacetime engineering. The highly specialized military skills frequently find no direct application in peaceful pursuits, and the graduates of the hard school of war have had to dispense with the acquisition of the basic scientific fundamentals. A further obstacle to the training of the younger generation of engineers is the draining away of technical instructors into the laboratories of industry. For the sake of industry and of national well-being these young engineers must not be neglected. Every educational agency that conserves the interest and furthers the knowledge of *technically minded young men,* at best a too-limited number, should be supported. Among such agencies, cooperative research holds a high rank.

A broader educational function of cooperative research is also important, to build up, within its field, commonly shared knowledge. The *height* of a building is determined by the *tallness of its spires;* its *stability* by the *breadth of its base.* Through pioneer inspiration, science pierces upward into the unknown; through the combined efforts of the many working on its practical utilization it constructs a *firm and enduring foundation of service to mankind.* The ancient sage said: "The race is not to the swift, nor the battle to the strong, neither yet bread to the wise, nor yet riches to men of skill; but time and chance happeneth to them all." Organized, consecutive effort, as in cooperative research, overcomes time and chance, and provides for a science or field of engineering a surer advance in the race of progress, and a *firmer stand* in the *battle* against the *obstacles to achievement.*

■ Multiplicity of Viewpoints

The value of cooperative research is immeasurably reinforced when to the force of numbers is added the advantage of representing a number of widely separated viewpoints: the joint undertaking of the automotive and petroleum industries manufacturing equipment, that of the petroleum industries refining the fuels and lubricants for automotive equipment, and that of the ultimate consumer who must purchase mutually adapted material from both the other two. The extent to which the automotive-petroleum research has thrived testifies to its utility. Industrial research as a whole has expanded *10 times* during the past 25 years; the rate of growth of the cooperative petroleum-automotive research has been, conservatively, *double this.*

An outstanding educational contribution of the CFR and the CRC has been to foster a conception of fuels and lubricants in the terms of the user rather than of the analyst. An analogy to this thought is the difference between the two conceptions of sound, as a product of vibrations of certain amplitudes and frequencies, and as a sensation striking the ear. A petroleum product may be identified by certain chemical characteristics and physical properties; it is a fuel or lubricant only in an operating engine, and its value as such should be measured under operating conditions. CFR, CER, and CLR projects have fostered this conception consistently.

Specific gravity was the first accepted criterion of gasoline quality. Volatility appeared of paramount importance in the early twenties, and was the subject of the first CFR project. Workers early pointed out that volatility might mean one thing to a refiner as a quality control for his product and quite another to the motorist as a factor in the starting, acceleration, and fuel consumption of his car. A volatility test was, therefore, developed simulating conditions related to engine operation – the equilibrium air distillation test. With the solution of the more pressing volatility problems, those involved in volatility too low for the then current usage conditions, interest in this charac-

teristic waned. Authors of a technical paper in 1935 dealt with it under the title, "A Forgotten Property of Gasoline."[1] More recently, interest in volatility has had a renaissance; equilibrium air distillation test procedures are being revamped, and data obtained by them being applied in studies of all types of internal-combustion fuel.

Meanwhile, the volatility pendulum had swung from the low to the high extreme, and, in 1929, the CFR accepted sponsorship for a comprehensive program of research on the vapor-lock problems so occasioned. The CFR studies made continuingly since that date have been pivoted on the view that vapor-locking characteristics are not definite fixed properties but phenomena to be investigated directly in connection with their occurrence in fuel systems.

With the entry of another main character, *detonation*, in the late 1920's, the drama of engine performance became definitely of the *"whodoneit"* type. Detonation posed the popular question, as seemingly unanswerable as the query on the priority of the hen and the egg: Which is responsible, the fuel or the engine? The CFR has maintained the view that knocking is a functional characteristic, to be judged by actual behavior. The effect on knock of fuel chemistry and of engine design has been investigated, but no widely accepted test has been based on such factors as criteria. Knock characteristics have been rated by performance, the comparison of the fuel or engine under test with the performance of or with standard reference fuels. For motor fuels, audibility has been adopted as the crucial performance feature. In other words, the knock characteristics of a fuel or an engine may be said to have been defined as its capacity for making *a noise*. In conducting aviation detonation studies, this criterion has not been very useful, because in an airplane in flight detonation is not heard under normal conditions. Cylinder-head temperature and various vibration pickups have been adopted as essential elements. An important current CRC project has as its objective the establishment of a common standard of knock intensity against which results obtained by various methods and equipments can be correlated.

The calorific content of fuels has recently come under critical review. A series of low-temperature volatility flight tests showed, even at relatively high carburetor air temperatures in the neighborhood of 80 to 85 F, a definite difference in minimum brake specific fuel consumption as between heavy and light fuels, even after they were corrected for Btu content. This gave rise to the thought that possibly an engine does not utilize the Btu content of a fuel in the same way as does the instrumentation used for measuring it. Further work to throw light on this aspect of Btu measurement is under consideration.

When in 1942 the joint research of the automotive and petroleum industries developed from the Cooperative Fuel Research Committee into the Coordinating Research Council, it added to its sphere the study of lubricants. Here also it has emphasized the concept of a material as a *functioning unity* instead of a *composite* to be identified by chemical or physical criteria. The current Gear Lubricants Groups were established on the direct request of the Army Ordnance Department with the ultimate objective of developing laboratory test procedures which would provide adequate evaluation of these lubricants for the Military Services. The Army did not report appreciable difficulty with the products meeting its then current specification, but it was not possible with that specification to procure the wide range of gear lubricants found commercially to give good service; the specification then existing was definitely restrictive. In response to the Ordnance Department's request, a series of test procedures for determining various operating characteristics was developed. The proposed new requirement will serve not only the Army, but civilian purchasers in securing suitable gear lubricants without restriction as to their nature and composition, and will open the door to further development work in both gears and gear lubricants, since the requirements are no longer tied down to the chemical composition of the product.

In the study of wheel bearing lubricants, the reverse conclusion was reached; existing specifications qualified as satisfactory greases that were not found satisfactory in service. A test apparatus and procedure were developed which yield results paralleling field behavior.

■ Fundamentals

The above discussion of the fundamentals of cooperative research can be summarized as follows:

The sphere of cooperative research lies between pure research in the outposts of science and industrial research applying the results of that scientific research in the development of commercial products; it embraces, too, the study of the *general* application of known scientific principles.

The important tools of cooperative research are those peculiar to it in its capacity for representing large numbers:

1. The statistical method.

2. The furtherance of understanding among technical men, which in its simplest form consists of the evaluation of a common vocabulary; and in its more advanced form, the development of yardsticks for measuring the properties and performance of materials and equipment, thus limiting and vitalizing existing technical terms.

3. Education, both in the sense of developing the understanding of less experienced men and providing a means of intercommunication among the more experienced, and the building up and preservation of a common body of knowledge.

Cooperative research that embraces a multiplicity of interests, such as CRC, has unique opportunity of contributing to technical advance, and an inescapable responsibility for so doing.

■ Basic Organization

For the organization of research, one fundamental has been established by scientists and administrators alike. This fundamental is that any such organization should not only *permit* but *promote* absolute freedom of technical thought and action. To quote a few statements of this principle:

Dr. Vannevar Bush, director of the Office of Scientific Research and Development: "All scientific progress results from the *free play* of *free intellects* working on subjects of their *own choice* in their *own way*."

President Truman: "Although science can be coordinated and encouraged, it cannot be *dictated to* or *regimented*. Science cannot progress unless founded on the *free intel-*

[1] See SAE Journal (Transactions), Vol. 37, August, 1935, pp. 293-300: "A Forgotten Property of Gasoline," by D. P. Barnard and J. O. Eisinger.

ligence of the *scientist.* I stress the fact that the Federal research agency here proposed should in no way impair that freedom."

Robert P. Patterson, Secretary of War: "Research and development in industry, as in the university, flourish best in an atmosphere of *complete freedom;* control will *wither science* by destroying its precious essence of originality and spontaneity. If I were to add to the four freedoms of the Atlantic Charter, I might suggest a fifth, *freedom of inquiry, experiment, and research.*"

Dr. C. E. K. Mees, vice president in charge of research, Eastman Kodak Co.: "No director who is any good ever really directs any research. What he does is to *protect the research men* from the people who *want to direct them* and who *don't know anything about it.*"

Freedom is a *prerequisite* for and *incentive* to *effective technical thought;* an important practical consideration reinforces further the desirability of self-government in technical research. As engineers advance in their commercial organizations, their duties become more largely concerned with administration and general policy-making, and less with purely technical matters. Hence, those whose position would entitle them to assume direction have lost intimate contact with the details of laboratory and experiment. In these times of rapid technical advance, such loss of contact, even for a short period, causes a noticeable gap in the knowledge needed for intelligent direction.

Freedom does not mean anarchy, in politics or in research. Cooperative research, lacking the incentive of individual gain, sometimes needs inspiration to spur a flagging interest, and guidance away from sterile or irrelevant fields. But *leadership from within,* not *direction from above or without* must furnish such inspiration and guidance.

In the CFR and the CRC the unit of organization has been the working Group, of from five to fifteen men, investigating a specific assignment. Every member, through interest, knowledge, and availability of facilities, actively contributes, and is expected to act on the basis of his technical convictions, rather than as an advocate of the policies of his company. Members of CRC Groups and Committees demonstrate their own convictions that they are contributing as individuals to a technical program by the extent to which they voluntarily devote their own time to planning and executing CRC projects.

In initiating a new project, a Leader may be appointed to organize the working Group; when the personnel of the Group already has volunteered, the Group elects its Leader. Such Leaders, possessing the confidence of their Groups, and the knowledge of their specific subjects, have so well lived up to their responsibilities that they and their Groups have been the keystones of CFR and CRC.

■ Supplementary Technical Organization

Any organization supplementing or superimposed upon the basic working units of a cooperative undertaking should have only one purpose, to serve these units. Such service is, first, technical and second, administrative.

The CRC committee organization permits only such ramifications as are required to provide the needed technical service. Only by constant vigilance and unrelenting effort can the number of Groups and Panels be kept to a minimum. A top-heavy mushrooming growth can be, and has been, prevented only by assigning every new project to an existing Group whenever there is such an appropriate Group, and, on the other hand, by persistently pruning out all dead branches by disbanding each Group promptly upon completion of its assignments and the rendering of a final report.

Working Groups engaged on allied topics are united under a Project Director, allied project collectives form Divisions, and Divisions are classed under one of three technical Committees, Fuel, Lubricants, or Equipment. Each step in the organization has its appropriate representative body, and Chairman or Leader.

In selecting the men throughout this organization it is imperative to observe the democratic principle of election by those to be represented. A six-year struggle between two schools of national thought has just ended. The first school believes in the equality of men under the law and that power over them should spring from the consent of the governed alone. The second school assigns to a selected race, and a selected class within that race, an inherent superiority, and upholds the validity of power invested in that class or race. The first school won, but the second was at times so near success as to arouse in certain quarters a suspicion that it might have something in its favor.

A cooperative technical organization is peculiarly suited to embody the democratic principles that for almost 200 years have been American, and within the next 200 years we pray may be universal. Its members are an ideal electorate, of more than average intelligence, with specific *interest in* and *knowledge of* the sphere to be governed. The observance of the democratic principle has been a bulwark of strength to the joint automotive and petroleum research. Young laboratory technicians, at first members of working Groups, have gone upward in its ranks as in the ranks of their individual companies. Having grappled firsthand with the problems of cooperative research, they are sympathetic and constructively helpful to those encountering such difficulties for the first time. Friendships formed during joint technical endeavor have been nourished to stalwart comradeship in Division and Committee duties. The resulting spirit of mutual respect and understanding is as substantial an ingredient in a cooperative undertaking as any more material factor. Pride in the organization they have created has inspired Leaders and workers to a high standard of technical integrity in its operation.

Many regard the committee system as essentially ineffective, and they are correct except under one all-important condition recognized by CFR many years ago, and recently delightfully expressed by Sir Arthur Salter in his book "Allied Shipping Control."

"*Nothing* is so *ineffective* as a committee which consists of persons *each of whom* has no *specialized function* and *no personal executive authority,* and yet tries to direct executive action. But if a number of persons, each of whom has a *direct executive authority* which he *continues to exercise in his own special sphere,* meet from time to time in order to dovetail their several measures and adjust them into a *common plan* and then return to their own tasks to put into effect what they have agreed, the *committee* is an *effective instrument* of *cooperative action.*"

CRC Groups meet the affirmative portion of this specification perfectly, as do also the meetings of Group Leaders held by Project Directors. Divisions and Committees

approach that degree of effectiveness only to the extent that their individual members remain active in personally and understandingly following the detailed problems of each Group.

■ Supplementary Technical Organization

To perform for the individual working groups the technical tasks they cannot encompass themselves is the function of the supplementary technical organization.

To prevent duplication of effort by working Groups is one such function exercised by CRC in two ways. First, Assignment Groups allocate topics for investigation. The most important of these are the Assignment Committee and the War Advisory Committee, the first of which acts as a liaison between agencies in industry and technical Groups and the second between the Armed Services and the same technical Groups. Second, central units handle specific projects for several interested Groups. For instance, the Aviation Fuels Division has an Instrument and Control Project to develop and coordinate the best practicable instrumentation and control apparatus for all variables encountered in testing aviation fuel performance. The Coordinating Lubricants Research Committee, and the CFR Motor and Aviation Fuels Divisions each have a Group to coordinate the requirements, supply, distribution, and the collection of pertinent composition and inspection data for lubricants or fuels, as the case may be, for all projects within their respective spheres.

A second function is to ensure that Groups obtain all information available in the cooperative organization to further their investigations. Meetings of Project Directors, Divisions, and Committees offer a *means of pooling information in reservoirs* from which all may obtain needed supplies.

A third function is to enlist the interest of widely divergent Groups for complex projects requiring coordinated action. For example, the CRC Group for the study of sludge formation has representatives of all three technical Committees, Fuel, Lubricants, and Equipment, providing for adequate investigation of every contributing element.

A fourth function is to ensure that material reaching an outside audience, either restricted reports or papers for general presentation, meets the technical standards developed over the years. In CRC, an analysis Panel, drawn from the Group conducting the research, generally prepares the report. Reports and papers are reviewed at Division and Committee meetings, and finally are approved for release by CRC Editorial Groups.

Finally, members in the upper ranks of the CRC supplementary organization are qualified by experience and their position in industrial, government, or educational institutions to present current and prospective cooperative undertakings to their respective organizations. In interpreting its aims and accomplishments, and in securing the participation of younger men in working Groups, they perform for CRC an *invaluable* and *valid* promotional service.

■ Central Headquarters Staff

The second type of service required by the working Groups is *administrative*. For this, a cooperative undertaking, like CRC, embracing some 300 units scattered over the entire country, needs an adequately staffed central office.

An important function of this central office is to relieve technical Groups of a maximum of clerical work, such as writing routine letters, scheduling meetings, preparing and circulating notices and minutes, and processing and distributing reports. A gage of the volume of such clerical work in 1945 is the 625,000-page copies of mimeographed material reproduced in the CRC office in the transmission of Committee reports. One project alone released 13 reports; a single series of tests required a 400-page report. The cost of this clerical work to industry is far less when performed by a carefully selected, trained, and experienced secretarial staff than when it consumes the time of relatively high-price technical men who are expert in the technical field involved but inexpert minute writers. Experience has shown that to encourage the utmost participation in laboratory time and directive thought, Leaders and Group members should be relieved of every possible item of nontechnical work, leaving each of them to do to the fullest that portion of the task for which he alone is qualified.

Minutes of meetings should record decisions made, and the reasons leading to them. They are historical source material, a current guide to active participants, and a medium of information for other interested and qualified persons. The preparation of minutes by a central office, instead of, alternatively, by a member of the Group, brings other benefits in addition to saving the time of technical men. An inexperienced man attempting to record a meeting cannot simultaneously function adequately as a contributing member of the Group. Further, his interest in the subject matter disqualifies him for impartially observing and recording discussions of controversial subjects, and usually results in one of two types of distortion. He may either unconsciously record his own familiar viewpoint too fully, or too conscientiously suppress his own valuable ideas, and thus give them inadequate weight. It is the common occurrence that a man who has devoted considerable time to a thorough preparation for a meeting ceases full activity in or withdraws entirely from the Group when he finds his arguments incorrectly or inadequately presented in the minutes of the meeting; and a continuation of such defects in minutes more quickly than any other factor defeats the purpose of a project.

Promptness in preparation of minutes and progress reports ranks only second to *quality*. "Immediate minutes" is the perfect answer, and the establishment of priority in the preparation of minutes requires judgment of a high order, since numerous conflicting and irreconcilable factors usually must be taken into account.

Always, and without exception, it is imperative that, immediately following each meeting, the central office know what has transpired in order to answer questions from all interested, particularly those invited but unable to attend. This is especially true when the minutes cannot be distributed promptly, and can be accomplished only when the staff has through one or more of its members a complete on-the-spot record of what occurred.

Further, meetings can be scheduled effectively only through a central office and certainly not individually by many scattered and isolated Groups. A central office can arrange concurrent meetings of interlocking Groups, and Group meetings in conjunction with other gatherings to

be attended by their members. Present "standing room only" transportation and hotel conditions make the consequent travel saving particularly welcome. Forty CRC Panel, Group, and Committee meetings and conferences took place in conjunction with the 1946 SAE Annual Meeting, and these had to be arranged so as not to conflict *with each other, with an even larger number of similar SAE meetings,* or *with attendance at the Society's technical sessions.*

Another duty of the CRC secretariat is to provide for the distribution of test material used in research programs. For example, during 1945, 211,582 gal of high-octane test and reference fuels were obtained and distributed for the aviation detonation programs, and 21,000 gal of fuel were made available to COT engine owners for use in aviation oil tests.

In performing its secretarial functions, a central office *coalesces the myriad separate units* of a widely scattered research undertaking into a *single, unified entity.* One expedient effective in accomplishing this has been to build up a *CRC code or designation system* for all *projects, test procedures, apparatus,* and *material* involved. This comprehensive, interlocking code, including provision for date of origin and character, provides an identifying badge and an historical capitulation for all CRC activities. It furnishes a reminder that no CRC project is an *isolated undertaking,* but part of a *diverse, harmonized program of research* adapted to the current needs of the automotive and petroleum industries.

The headquarters staff also styles the processed reports uniformly, so that, in form as well as in content, they represent the organization. More important, the central secretariat is a repository for records and reports. This contributes toward preventing the organization from resembling Mark Twain's mule, *"with no pride of ancestry or hope for posterity."* An historical background is built up and preserved; and a store house of technical information is accumulated, upon which the organization may draw in the future.

Finally, financial support and general administration must be provided. This is the primary province of a Board of Directors, on which the Society of Automotive Engineers and the American Petroleum Institute are equally represented. The major part of financial support comes from equal appropriations of the two organizations. In addition, certain branches of the Armed Services and closely related sections of industry bear the direct expense of specific projects which they have requested or in which they have a special interest. Such contributions are accepted on the same basis as money paid in at a railway or theatre ticket office. The prospective purchaser may buy or refuse to buy, but paying money for a ticket does not bring with it the right to *dictate individually* the *conditions of his ride* or the details of the *artists' performance,* nor does it guarantee the donor's *personal satisfaction with the results.* Such projects, like all CRC tasks, are guided by *technical considerations,* of which the best judges are the *men actually doing the work.*

This, in effect, is the theme of this discussion. In an analysis of the expression, cooperative research organization, the conclusion must be reached that it is *results in research* that justify the organization. No matter how *loftily* or how *exactly aims are stated,* or how *elaborate the organization,* all is *form, devoid of content,* without the *technical conscience, intelligence,* and *skill* of the man in the laboratory. An apt example is the decline in precision of knock ratings of the CFR exchange tests during the war period. The test procedures had been worked out carefully, had received the usual CRC approval, and had been applied successfully over a period of years. However, when the competent, experienced operators were called away by the exigencies of war, actual test results deteriorated.

A second conclusion to be drawn from an analysis of the expression, cooperative research organization, is the importance that must be assigned to the word *"cooperation."* At no previous point in the world's history has there been so strong a trend as prevails today toward coordinated effort as the one means of survival for individuals, industries, and nations. In the technical field, this is exemplified by the recent statement of a prominent member of one of this country's foremost research organizations that his company, as a matter of policy, planned more than ever before to participate actively in technical programs such as those of the SAE and the CRC because only such cooperative activity could ensure that its immense development organization was in step with *current and prospective industrial progress.*

More than a quarter of a century ago *Harry Horning* appreciated the value of *technical cooperation.* He conceived of, and, as a natural leader of a small group of men of sympathetic judgment and vision, set about establishing a *broadly cooperative agency* devoted to the *better mutual adaptation,* through experimental effort, of motor fuels to internal-combustion engines and engines to fuels. Later he accepted the chairmanship, and, against stubborn opposition in some quarters, organized in the SAE Research Department the Society's first committee on engine oil research. This *initial effort matured* into the present Coordinating Lubricants Research Committee.

He took a leading part in transferring the Society's research on *aviation* and *automotive diesel fuels* to the CFR Committee, thus bringing into being the Committee's Aviation Fuels and Automotive Diesel Fuels Divisions. He was the *first chairman* of the CFR Group studying *alcohol fuels,* and blends, and continued in this position until his death. From this Group was developed the Non-Petroleum Fuels Division of the CFR, of which B. B. Bachman, President of the Coordinating Research Council, Inc., is now Chairman. Thus the *man so fittingly memorialized in this award,* more than any other single person, was truly the *father* of the *automotive interest now centering* in CRC.

We of his associates who are privileged to continue actively in this cooperative enterprise all owe him a debt of deep gratitude, and have a pleasant but serious obligation to meet, not only by each doing his part in current projects, but by inspiring the younger men, the real workers, to take full advantage of their cooperative capabilities and opportunities to carry on the work started by Harry Horning and other notable CFR founders.

To this *end, and in conclusion,* with the full realization that in cooperative research, practical *research results is the only end that justifies an organization,* let us not forget that all aspects of the organization, Administrative Boards, Technical Committees, Divisions, Project Directors, and Secretariat must serve those who do the actual research, the *Working Groups and their Leaders.*

FLEXIBLE or SPRING MEDIUM o S

THE subject defined by the title of this paper is so large that we shall only treat some of the problems of suspension spring design, particularly those that have been of major interest in recent years.

■ Spring Rate

Strictly speaking, any member in an automobile chassis is flexible, and many operating troubles arise because the designer has overlooked that fact. However, there is good reason for singling out *the* spring and treating it as *the* flexible member, because flexibility, and not rigidity, is its prime characteristic.

The basic action of this flexibility is best seen on a load-deflection diagram such as Fig. 1, which shows loads at the wheel against deflections at the wheel.

At any load, spring rate equals change of load divided by change of deflection. The slope of the tangent to the load-deflection curve has this value.

For indicating the softness of a spring as part of a suspension, it has become common practice to use the value of static deflection. This equals: static load divided by rate at static load; it is illustrated in Fig. 1.

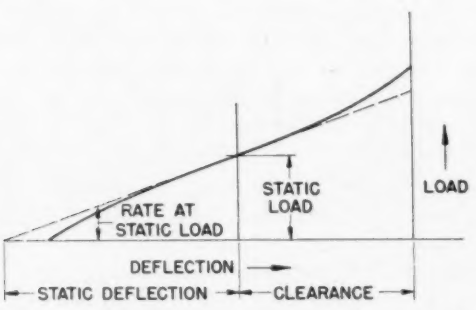
The distance which the wheel can be deflected above static deflection is usually called clearance or ride clearance.

The area under the curve, between zero load and maximum compression, is proportional to the energy to be stored in the spring. If the load-deflection line is straight, this energy is determined once the static load, static deflection, and clearance are given. For a given load and clearance, it is smallest if static deflection equals clearance.

For most passenger-car suspension springs the static deflection is larger than the clearance, and in this case a softer spring must store more energy and hence be heavier than a stiffer spring.

On many suspensions the rate is increased, either by distortion of the main spring or by linkage members, such as rubber bushings, which add to the rate but do not carry load at the static position. The main spring then must be made softer and heavier in order to produce a given total rate. Fig. 2 illustrates such a case. For best economy of material, all elastic parts should carry load in proportion to their rate.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 9, 1946.]



■ Fig. 1 – Load-deflection diagram

■ Stored Energy

The amount of energy that a spring can store per pound of spring weight is proportional to the square of the highest permissible stress times a certain numerical factor, depending upon the uniformity of stress, and inversely proportional to the modulus of the material.

Commercial springs, as used in suspensions, can store the amounts of energy per pound of total spring weight shown in Table 1.

These figures show spring efficiency in terms of spring weight only; they do not indicate economy of space or of total weight of mechanism. It would, therefore, be a mistake to rate springs by this factor alone. But the figures do indicate in which types the weight economy needs the most improvement.

■ Variable Rate

The load-deflection curve is usually not a straight line, but curved as shown in Fig. 1. Rubber in shear has a nonlinear stress-strain relation, which fits the requirements of a suspension very well. Rubber used in compression usually produces a strongly nonlinear spring because of distortions and changes of effective area during deflection.

Air, if used with a constant piston area, produces a rapid increase of rate with deflection. If a bellows is used as container, the effective area will vary during deflection. This fact makes it possible to control the rate increase by the design of the bellows.

Most steel springs in themselves have a practically uniform rate, but the linkage often changes this to a variable rate at the wheel.

When it is desired to produce a variable rate, this can be done either by controlling the geometry of the linkage or by making a spring with inherently variable rate.

Most variable rate springs are not economical of material because the maximum stresses vary greatly between various parts of the spring, but there are many ways to produce a variable rate without sacrificing economy. To achieve this, the maximum stresses in all sections of the spring must be kept uniform.

Very low rates can be produced by a toggle mechanism, as is shown in Fig. 3. This consists of a stiff main spring and a toggle. This mechanism produces a low rate near the static position, with a small amount of spring steel.

The belleville spring is really an example of a toggle mechanism. The main spring and toggle spring are simply represented by different stress patterns within the same

SUSPENSIONS

by
ROBERT SCHILLING
General Motors Corp.

PROBLEMS of suspension spring design that have been of major interest in recent years, such as spring life and reliability, hardness and settling, fatigue, shot peening, and presetting, are presented here.

Various types of springs are discussed by Mr. Schilling, who concludes that no one type is superior in all respects, hence commercial applications of different types have remained competitive.

He feels that this picture will not change in the future, that is, no one type will be used exclusively. Considering the great superiority of some types in economy of material, however, he expects that the less economical springs will gradually be replaced unless their weight can be substantially reduced.

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part. The variety of load-deflection curves obtainable is shown in Fig. 4. The disadvantage of this spring is that very high dimensional accuracy is required to produce low rates accurately. A comparison of weight economy with other springs is, therefore, only possible after production tolerances have been established.

Another arrangement that forms a toggle mechanism is the combination of leaf spring and shackle. Fig. 5 shows how the shackle may produce either a compressive or tensile force in the spring itself and therefore make it into a toggle. This action causes large variations in rate which can be predicted.[1]

These last examples could also be classified as rate variations due to the linkage, although generally we mean by that a linkage in which the ratio between wheel and spring is varied. Practically all linkages used in practice are of this nature, and therefore practically all suspensions show a variable rate.

Patents and articles in the literature disclose many supposedly novel ways by which variable rates may be achieved, but we don't need novel arrangements, we only need to proportion properly the mechanism now in use. The practical problem is really at which points of the load-deflection curve we shall increase the rate, and by how much.

Rate buildup is desirable for several reasons:

1. It softens striking through. For this purpose the rate should increase during the compression stroke, especially near the end of that stroke. The rate increase required is so large that it cannot usually be obtained by the main spring alone without excessive loss of efficiency. A separate spring is usually added for this purpose in the form of a rubber bumper.

[1] See "Manual of Design and Application of Leaf Springs," SAE War Engineering Board, October, 1945. Published by SAE Special Publications Department, 29 West 39 St., New York City 18, for $1.00 to members and $2.00 to nonmembers.

Table 1 – Energy That Can Be Stored by Commercial Springs

	Energy per Lb of Spring, in.-lb
Leaf Springs	300 – 450
Helical, Round Wire, Coil Springs	700 – 1100
Torsion Bar Springs	1000 – 1500
Volute Springs	500 – 1000
Rubber Spring in Shear	2000 – 4000

Table 2 – Spring Example

	Load = 100%	Rate Constant	Load = 140% Static Deflection Constant
Static Load, %	100	140	140
Rate at Static Load, %	100	100	140
Static Deflection, in.	8	11.2	8
Deflection from 100% Load to 140% Load, in.	—	3.2	2.7

2. If the rate is increased in proportion to the load, the suspension has the same static deflection under all loads, and hence the same basic frequency. The ride will then not change under different loadings of the car.

But, rapid change of rate near the static load position makes for a jerky ride, so, for comfort, while continuously increasing rate is desirable, it should not increase as fast as the load increases.

3. Rate buildup in compression will reduce the change of standing height under different loads. In order to get this result, the rate increase again must be continuous and the same objection exists as under (2). Due to this limitation on rate buildup, the change of standing height can be reduced only a little. Consider, for example, a suspension with 8 in. of static deflection and a static load increase of 40%, as given in Table 2.

In this example the change of standing height is reduced by only 15%. The corresponding load-deflection diagram is shown in Fig. 6.

At the risk of starting some arguments, I have drawn in Fig. 7 what I consider a satisfactory load-deflection diagram for passenger cars that are to run on average U.S.A. roads.

The specification would run:

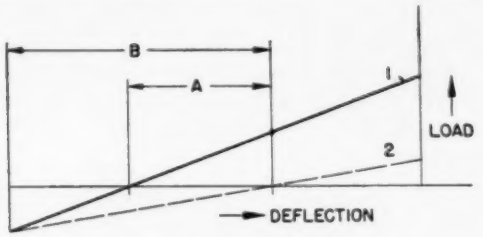
Static Deflection – 6-12 in., depending on size of car.

Clearance – 4-6 in., depending on static deflection.

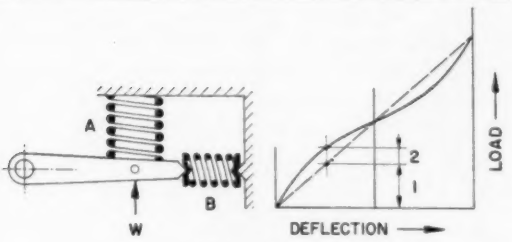
Rate Increase – To be no more than 20% at a deflection of 60% clearance above and below normal standing height. Below this range it may build up faster; above it, it must increase rapidly but smoothly.

■ Life and Reliability

Springs are much more carefully designed and subject to more laboratory development than other parts of the chassis for two reasons:



■ Fig. 2 – Effect of parasitic rate
1 – Desired total load-deflection curve
2 – Parasitic forces in linkage
A – Desired static deflection
B – Static deflection required for spring



■ Fig. 3 – Toggle suspension
1 – Load from main spring
2 – Load from toggle
A – Main spring
B – Toggle spring

Ideally, the amount of material required varies inversely as the square of the operating stress. This means that considerable savings are possible if the operating stress can be raised.

The operating loads and deflections are known much more accurately than for any other part, and overloading cannot occur. A reliable analysis is, therefore, possible.

These efforts, however, would be useless if we could not rely on the uniformity of the final product in respect to quality of material and processing. If only one part in a hundred had considerably poorer life than the average, we would have to base our design on this poorest part. For this reason, techniques which improve the uniformity of the product have received much attention. It is very much to the credit of spring manufacturers that spring breakages on passenger cars have practically disappeared, although we use higher operating stresses than 10 and 20 years ago.

The operating stresses are limited by two factors: Settling and fatigue fracture. This is true for any material, but if we now start to discuss the relation of the two, we must consider one material at a time.

■ Hardness and Settling

For steel, settling depends mainly on the relation between highest operating stress and yield point. The stress range during operation has, if any, a minor influence. From this standpoint, then, it becomes desirable to use the material at the highest possible hardness.

The limit of usable hardness is set not so much by average fatigue properties but by reliability, and reliability depends not only on the steel itself but very much upon the surface of the part. It is, therefore, extremely important that a spring have not only a smooth surface but one that is free from decarburization and other defects that may be caused by processing.

It has been demonstrated that steel at hardnesses far above usual practice has excellent fatigue properties, but in this state it is very sensitive to stress concentrations, especially those produced by inclusions. For commercial products the hardness must, therefore, be held to values where these stress concentrations can relieve themselves by local settling without causing cracks. If in the future we can rely on getting steel with fewer and smaller inclusions, a higher hardness may become usable.

■ Fatigue

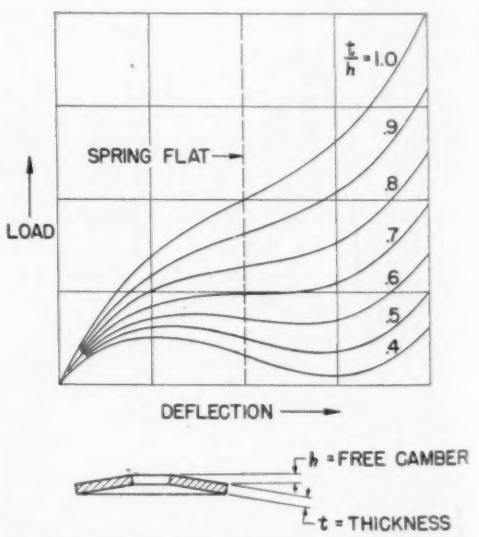
The fatigue life of a part depends both on the maximum stress and on the stress range. But in the service encountered by passenger-car springs in the U.S.A., the many small deflection cycles do far less damage to the material than the few large ones. It has been found that a spring will have satisfactory life on the road if laboratory fatigue tests show an average life of 100,000 cycles from a small preload to maximum service load. This is only true for operation on our average roads. Vehicles which operate cross-country or on very primitive roads may break such springs in a few thousand miles.

The ideal spring steel will be one which gives satisfactory fatigue life when stressed up to the settling point. Our spring steels just about meet that specification for use on passenger cars, partly due to luck, and partly due to the care exercised in production.

Special credit should be given to some processes which have improved both the average fatigue life and reliability.

■ Shot Peening

First among these is shot peening. The American spring industry was one of the first to use this process on commercial products and has done a great deal of pioneering work. Fig. 8 is based on data which were furnished by Hoy Clark from tests by Eaton Mfg. Co. It shows some examples of the increase in life produced by shot peening. Although this is only a small sample of the voluminous data on this subject, it is typical in showing the consistent improvement at any operating stress.



■ Fig. 4 – Load-deflection curves for belleville springs

Since these data are based on a comparatively small number of samples, they cannot show the improvement of reliability. The practical value of shot peening is, therefore, even higher than is indicated by this illustration.

It is important to design parts so that they can be shot peened. The volute spring, for example, has many desirable characteristics as a suspension spring. It is compact and can be designed to produce a suitable rate increase; but, just because of its compactness, it cannot be shot peened and therefore will never be as reliable at high operating stresses as are other types. Another example is the torsion bar spring with splined ends. In this part, high stress concentrations at the spline roots are unavoidable, but good service life and reliability can be obtained if the splines are shaped so that they may be effectively shot peened.

Surface defects are produced not only in processing but also in service, for example corrosion, fretting, and scratching. Shot peening is also very effective in preventing damage from these.

■ Presetting

Another process which lately has received much attention is presetting. It is really a very old trick and has been used for generations, but only recently have we realized how much can be gained by it.

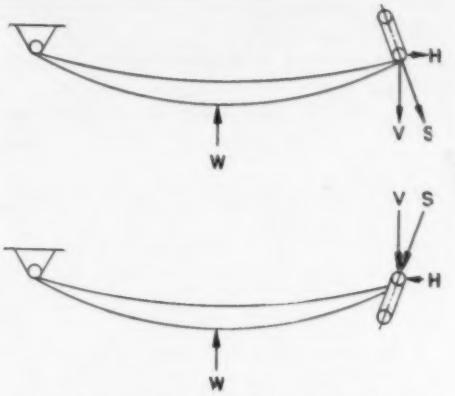
Its effect is quite obvious. When a part is being loaded, some portions will reach their yield stress long before others; either because they are subject to stress concentration from fillets or surface defects and inclusions, or because they are highly stressed due to the normal stress distribution. These parts will then yield during the first load application, and at maximum load will just be loaded to the yield stress. This means that the part will take a set during this first load application.

If the service load is the same as the presetting load, the settling in service will be much reduced. If the service load is less than the presetting load, no portion of the material will ever again be loaded up to its yield stress, so that settling becomes quite small and fatigue life is improved. For best results, the spring should be designed so that it can be preset past the operating load. This is always possible for torsion bar springs, leaf springs, volute springs, and belleville springs. Springs with limiting solid height, such as helical coil springs and ring springs, should be designed so that they are not compressed solidly in service.

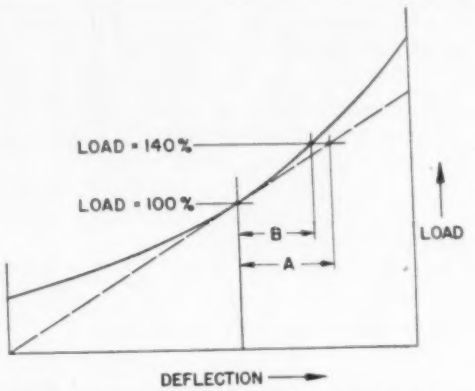
How much presetting can increase the capacity of a spring is shown in Fig. 9 for a torsion bar spring, which is the simplest example. The solid line 0-1-2 is the load-deflection curve of a bar during the presetting operation. The scales to the left show the torque applied, those on the right show nominal stress, calculated as torque divided by section modulus.

At point 1 the stress at the outermost fibers has reached 110,000 psi, the yield stress of the material. From then on the bar begins to yield, at the outer fibers first and progressively toward the center as the windup increases. The torque therefore increases more and more slowly. From this curve the stress at the outer fiber can be calculated at any stage.[2] It is shown as curve 1-3. This curve shows a gradual increase of stress at this point, due to increasing yield strength caused by work hardening.

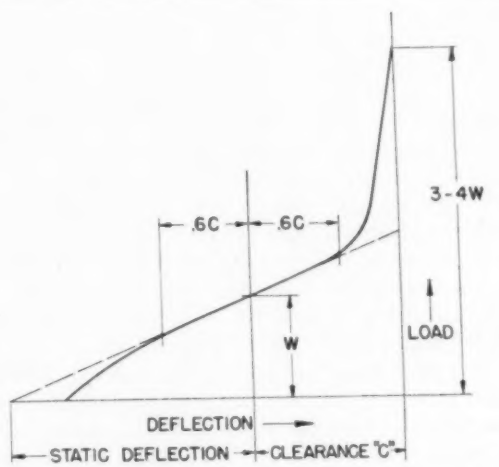
[2] See "Materials of Construction," by G. B. Upton. Published by John Wiley & Sons, New York, 1916 (out of print).



■ Fig. 5 – Toggle forces in leaf springs
S – Total shackle force
V – Vertical component of shackle force
H – Horizontal component of shackle force producing toggle effect

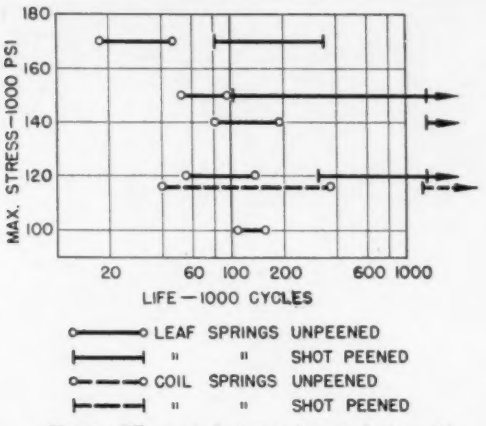


■ Fig. 6 – Change of standing height with variable rate
- - - - Constant rate
—— Constant static deflection
A – Change of standing height with constant rate
B – Change of standing height with constant static deflection



■ Fig. 7 – Load-deflection diagram for passenger cars

The line 0-1-3 is then actually the stress-strain curve of the material, and also shows the stress distribution along one radius, from center to surface, under the highest presetting load. The distance scale in per cent of radius is,



■ Fig. 8 – Effect of shot peening on fatigue life

therefore, also marked along the bottom of the diagram.

If the bar carried the maximum torque elastically, the surface stress would have to be 152,000 psi, as indicated by point 2, and the stress at any point along a radius would then be indicated by the straight line 0-2. If the load is removed, the bar unwinds elastically, and all stresses decrease by an amount corresponding to the ordinates of line 0-2. The difference between line 0-2 and line 0-1-3 will then remain as trapped stress. Consequently the load stress or stress range is equal to the nominal stress, but the maximum total stress is given by line 0-1-3.

The process of cold setting is usable for all springs or any structural parts which are loaded in one direction only, whether the stresses are shear or bending. It may well be used on suspension parts, such as axles or wheel arms, where settling under load is a limiting design factor.

The cold straightening of inaccurate or damaged parts is really the same process, and will leave trapped stresses. If such cold straightening is to be used on structural parts, we must, therefore, take care to see that the trapped stresses are favorable. This requires that the last straightening operation be in the same direction as the normal load deflection.

Presetting will produce the greatest increase of load capacity in parts with the most irregular stress distribution, beeause its success is due to the equalization of maximum stresses. For example, it is more effective on rectangular wire coil springs than on round wire coil springs, but it is of little use on specimens under uniform tension or shear.

There is, of course, a limit to which the process may be carried, for under excessive presetting the material may be damaged and its fatigue life impaired. This danger is greater when the parts are under a simple tension stress, and for this reason presetting of bending specimens has not given as good results as on shear specimens.

On the other hand, the rate of load increase during progressive presetting becomes less and less, so that there is not much use in carrying on past a certain stage. On round wire springs this point is reached when the yielding has progressed to about one-half the radius. This will have increased the load capacity approximately 40%. That means the nominal stress is then 1.4 times yield stress and the stored energy is double that carried at original yield stress.

[3] See SAE Journal (Transactions), Vol. 49, October, 1941, pp. 442-447: "Static Fatigue Life of Rubber," by S. M. Caldwell, R. A. Merrill, C. M. Sloman, and F. L. Yost.

The fact that presetting is so much more effective on round wire in shear than on strips in bending accounts for a great deal of the difference in efficiency between coil springs and leaf springs. Commercial practice is shown in Table 3.

This table indicates that much improvement can still be made in leaf springs.

■ Elastic Properties of Rubber

Rubber as an elastic material has quite different characteristics. We shall not discuss here such phenomena as aging and changes due to temperature, because they vary widely among compounds and we always hope that the rubber compounders will be able to reduce them to unimportant magnitudes.

A serious limitation to the use of rubber for load carrying elastic members is its settling. In contrast to steel, rubber does not settle only when the load exceeds a certain yield stress, but it settles even under low stresses. This settling is moderate under small stresses and becomes impossibly large only above a certain limiting stress – which depends on the compound used. For soft compounds, suitable for elastic members, this is in the neighborhood of 150 psi. But even the small settling below this stress continues for months and possibly never stops completely. It has even been demonstrated that rubber will suffer from static fatigue; that means it will keep settling and finally break under small stresses.[3] Furthermore, the settling depends on temperature and exercise. The standing height of a car on rubber springs will vary with temperature and will be different after it has been stored than after it has been used.

For a suspension using rubber springs an automatic adjustment of standing height is, therefore, highly desirable. If this is not used, the stresses must be kept low enough so that the settling and variation of standing height stay within commercially acceptable limits. Hence, this is the starting point in the design of a rubber spring.

When a rubber spring is designed for acceptable settling, the fatigue life does not seem to be a problem. Fatigue properties of rubber are good, especially when run under a preload. However, rubber is also sensitive to stress concentrations, just as is steel. In a way this is a more serious problem here, because rubber does not have steel's ability to yield under overload and in this way to reduce excessive stresses. Further, under the high strains used with rubber, local buckling and folding can occur, which will produce stress concentrations that are hard to anticipate.

The main fatigue problem in rubber usually appears in connection with the bond. Nonbonded rubber parts are not suitable for highly loaded springs, because it seems impossible to hold them solidly at the mounting surfaces and to prevent chafing and abrasion.

The ultimate strength and the fatigue strength of the bond or near the bond are lower than that of the rubber itself, so that the shear stress at the bond must be limited

Table 3 – Commercial Practice

	Leaf Spring	Coil Spring
Yield Point of Material in Tension, psi	190,000	.....
Yield Point of Material in Shear, psi	.....	115,000
Nominal Operating Stress Including Curvature Corrections, psi	150,000	150,000
Ratio of Operating Stress to Yield Point	0.80	1.3

to a maximum operating value of about 200 psi. At the bond the rubber is restrained in two directions, a short distance away from it, it may deform in all directions, consequently stress concentrations are almost sure to exist in the neighborhood of the bond, especially along the edges. It is here that fatigue failures are most likely.

In rubber as in steel there are means to produce trapped stresses, so that we may expect even better fatigue properties in the future.

Although it is difficult to make a satisfactory suspension spring of rubber, this material is well suited for bump springs. Here it does not carry a static load, so that settling is absent. For the few full load cycles required in service, probably less than 10,000, rubber may be very highly stressed. Failure of the bond is usually prevented by putting it under compressive stress. Almost any desired load-deflection curve can be produced by controlling the shape of the rubber block and of the contacting surfaces.

Air is really the ideal elastic medium. It does not settle and it does not fatigue. Material problems, however, appear in the design of a container.

■ Damping in Springs

This paper is not the place to discuss damping characteristics of springs in detail but a short expression of opinion should be permissible.

Damping in steel springs is of the variety which we call static friction. That means there is a certain friction at zero velocity which may increase, but usually decreases with velocity. This friction is useful for damping in cross-country vehicles, but it is a great handicap to the ride engineer who wants to produce a smooth boulevard ride. Many attempts have been made to control the friction, so that it is more uniform and causes less sticking but even then, any spring that possesses it is under a disadvantage when competing with frictionless springs on a passenger car.

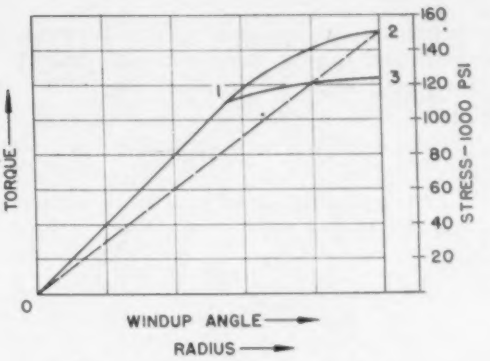
Damping in rubber springs is of a completely different nature. It is usually classed as hysteresis. That means, a load-deflection diagram, run slowly, will show a big frictional loop due to the settling of the material. If the test is run quickly this loop will be very small, because the material has had no time to settle. The deflection cycles in a suspension are fast enough so that this type of damping is zero for practical purposes.

Air as an elastic medium has practically no damping in itself, but it is possible to provide restrictions in the air passages so that considerable damping can be produced. However, provision must be made that the air is forced positively back and forth through these restrictions, even for high oscillation frequencies and short strokes, otherwise the damping is ineffective under these conditions.

■ Other Duties of Springs

When assessing the value of various spring types we must also consider that they may be used for purposes other than to support the load elastically.

Leaf springs often form the whole linkage by which the axles or wheels are attached to the chassis. In this way they save the expense and weight of structural members and thus can compete with much lighter and cheaper springs of different types. If, however, even with leaf springs, radius arms and links are necessary to guide the wheels, the leaf spring loses one of its main advantages. Hence, when designing a leaf spring suspension, it should be treated as a link as well as a spring.



■ Fig. 9 – Stresses in preset torsion bar

The volute spring inherently has a rapid rate increase toward the end of the stroke. If it is used as main suspension spring, a separate bump spring is not necessary. However, excessive rate increase causes unequal stress distribution and may make the spring uneconomical, hence it should be properly proportioned.[4]

Coil springs are only used to support the load, but in practically all installations they are forced to deflect laterally and to twist during compression. These deflections produce forces and moments[5] which usually add to the rate of the suspension. Such additional stiffness is uneconomical, as already shown, besides increasing stresses, and must be kept small.

Rubber springs, when used in the form of bushings, can replace the main suspension bearings and therefore not only save parts but also eliminate the need for lubrication. Besides this, their flexibility in all directions gives some insulation against shock and noise.[6]

Some springs have disadvantages in this respect, because they require expensive methods of fastening.

While the coil springs and volute springs need nothing more for support than a few simple surfaces in some sheet metal part or on the frame, torsion bar springs must have separate anchors which are highly stressed, leaf springs must use shackles, a rubber-air spring needs reservoirs.

The economical value of a certain spring type can, therefore, not be taken in the abstract, but must be investigated for the installation as a whole.

■ Space Requirements

Another point, which often decides in the choice of a particular spring type, is the availability of space.

Ideally a spring should be placed directly between the wheel or axle and the load, because in this way the joints of the linkage are not loaded statically. This occurs naturally with leaf spring installations such as the Hotchkiss drive, with the additional advantage that the support is spread to several points on the chassis. But a good leaf spring must be fairly long, and many leaf spring applica-

[4] See "Manual of Design and Manufacture of Volute Springs," SAE War Engineering Board, May, 1945. Published by SAE Special Publications Department, 29 West 39 St., New York City 18, for $1.00 to members and $2.00 to nonmembers.

[5] See p. 169 of "Mechanical Springs," by A. M. Wahl. Published by Penton Publishing Co., Cleveland, Ohio, 1944.

[6] See SAE Journal (Transactions), Vol. 45, November, 1939, pp. 471-477: "Rubber Suspension," by A. S. Krotz.

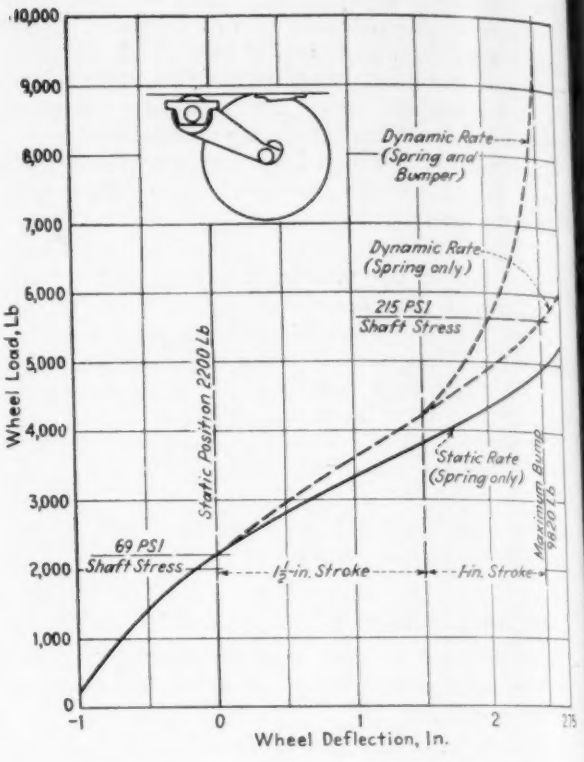
tions, notably on front axles, have been ruined by allowing too little length.

The volute springs, ring springs, and belleville springs are very compact but not suitable for long strokes. They must, therefore, be located at some point in the linkage where the stroke is reduced.

Coil springs are a little less compact but can be designed equally well for long or short strokes. However, if they are placed directly above the wheel or axle, they must have a long stroke and hence require height that is not always available.

Torsion bar springs have little volume, but all their volume is stretched out along the centerline. The length increases with the required windup angle. To make the bar short, the wheel arm must be made long, and this produces higher torque and hence more expensive fastenings.

Summarizing our analysis, we see that no one spring type is superior in all respects. Hence commercial applications of different types have remained competitive. This picture will probably not change in the future; I don't expect to see one type become used exclusively. However, considering the great superiority of some types in economy of material, the suspension designer's aim will always be to use those types and develop suitable applications for them. The more we learn about the geometric and dynamic requirements of suspensions, the easier it will be for us to fit any type of spring, hence the less economical springs will gradually be replaced unless their weight can be substantially reduced.



■ Fig. A – LVT tank suspension – torsilastic spring

■ ■

DISCUSSION

Torsilastic Spring Used on LVT Tanks

– A. S. KROTZ
B. F. Goodrich Co.

THERE can be no doubt that as a spring medium there is nothing to compare with air. Some excellent work has been done in applying such springs. By comparison, rubber probably should occupy an intermediate position and steel would be least desirable. However, the application of some types of rubber springs can be made more simply than air or steel, and it is probable that on most suspension designs the selection of one of these three materials will depend, primarily at least, as much on the ingenuity of the designer in applying the spring as it will on the efficiency of the material or the inherent virtues of the various types of springs.

We at Goodrich believe there is a field for several diverse kinds of steel, air, and rubber springs and that it is a healthy sign in the industry that the engineers responsible for the technical features of their product (with the forbearance of the sales department) are selecting a wider range of springs than in the past.

In general, the life of rubber springs is determined more by fatigue range than by mean static stress. This is illustrated by two load deflection curves of rubber torsion type of production springs which are included here, although they are not passenger-car springs, but with the thought that they provide background on rubber springs which would be applicable to passenger-car design work. The first one marked Fig. A shows the suspension spring of the LVT tank. We made approximately 500,000 of these springs and the suspension was very satisfactory. The static shear stress on the rubber, where it is bonded to the shaft, was 69 psi. At bump position, the springs stored 4877 in.-lb of energy per lb of rubber. The life required of the spring was relatively short and it was, therefore, possible to use a very large fatigue range. Over some types of terrain, the rubber stress at the shaft probably varied every few seconds from some place near 0 to over 200 psi. With bumpers, the maximum vertical bump load at the wheel was approximately 10,000 lb or five times static load.

Fig. B shows the load-deflection curve for the postwar suspension now in production on the 44-passenger Twin Coach bus. In this case, the maximum static shear on the bond between the rubber and the shaft for standee condition (figured at 200% of seated load) is 132 psi. Maximum shear on the rubber at bump position is 180 psi. At this position, the spring stores 2800 in.-lb energy per lb of rubber compound. Laboratory tests as well as road tests indicate that we may expect one-half million miles or more life from this spring. This is many times the life expected of the tank spring illustrated first, and the increase in life is obtained in spite of greater static stress by a reduction in fatigue range, and of course, by the smaller number of large fatigue cycles in a given period of operation.

The loading curve shown in Fig. B illustrates an application where a variable rate proved very useful. While the load-deflection curve of the torsion spring itself is almost a straightline through the useful range, linkage was provided to change the effective arm length, with the result that with very little sacrifice in ride, the change in floor level from empty to standee load was diminished by 1.17 in., as shown at *A* on the curve. This was extremely important in a design where there was considerable change in load and where maximum wheel travel from empty to bump was limited to less than 5 in.

The curve shown as Fig. C indicates the change in nominal dynamic deflection for this suspension and illustrates the point made by Mr. Schilling that with increasing loads, the spring rate should not increase as fast as the load in order to provide satisfactory ride. The overall ride gives the impression of considerably softer springs, particularly on boulevard ride, due largely to the elimination of dry friction effect and the reduction in harshness and noise level.

We have obtained very encouraging results with springs designed for considerably higher static stress and bump stress than shown on these curves. We have some test installations of torsion-type rubber springs where static stress is as high as 200 psi shear at the bond to the shaft.

Where rubber springs or mountings are tested to destruction, the

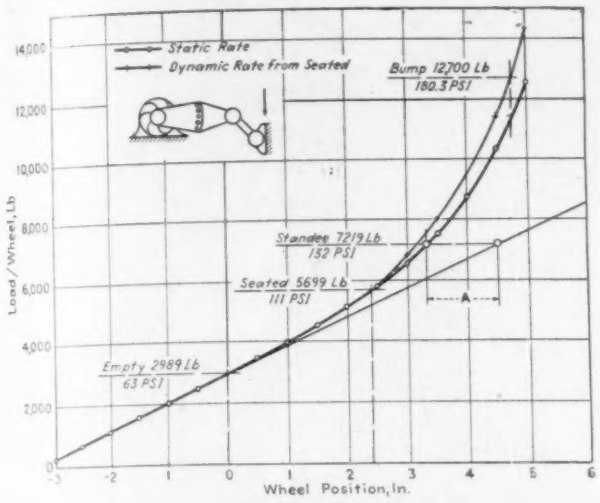


Fig. B—Twin Coach bus suspension—44 passenger—rear torsilastic spring

type of failure gives some indication of the degree of success the designer achieved in avoiding distress areas. A flat bonded rubber sandwich will usually fail at the bond in the tension areas at the ends. The torsion type of rubber spring can be thought of as such a sandwich wrapped around to form a continuous cylinder. The distress areas that were formerly the ends then disappear. While it is sometimes impossible to get more than 150 psi shear on a flat-plate sandwich, we can stress many types of cylindrical springs well over 500 psi in the laboratory, although as yet we do not recommend such stresses in actual installations. When failure does occur on such a cylindrical torsion spring, it is seldom at the bond at the inner diameter of the rubber, even though the stress is highest there.

As pointed out by Mr. Schilling, the energy stored in unit weight of a spring varies as the square of the stress. While it is not always so important with steel springs, it is very important in rubber springs to recognize that where the fatigue range is a fixed *per cent* of the static stress, the magnitude of fatigue stress range varies directly as the static stress; and the flow of energy in the spring material varies as the square of the static stress.

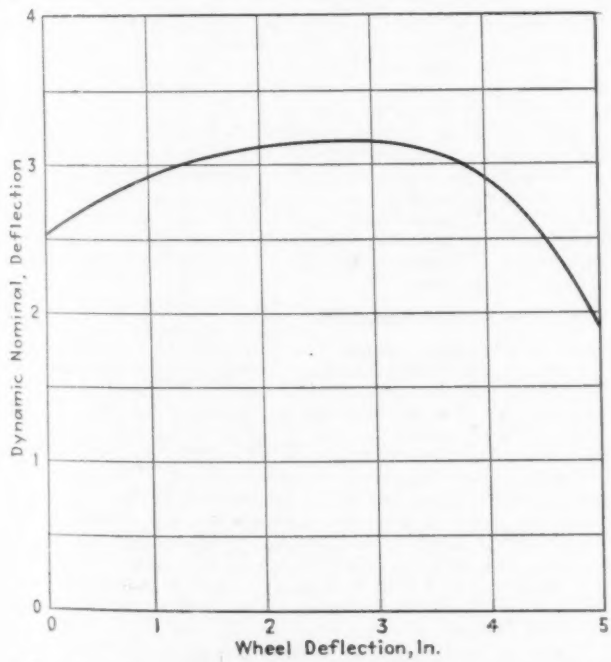


Fig. C—Dynamic nominal deflection for 44-passenger Twin Coach bus suspension—rear torsilastic spring

It is well known that where spring rate is reduced, it is usually necessary to increase the amplitude of wheel stroke. General practice seems to indicate that wheel stroke or breathing clearances need not be increased by as large a percentage as the change in spring rate. It, therefore, follows that springs used in suspensions having a soft rate can logically be expected to carry higher static stress than those used on stiffer suspensions. This makes no allowance for those special problems such as added cocking of the bottom coils of a coil spring if the spring rate is reduced and greater clearance is allowed without lengthening the wheel support arms.

The question of optimum hardness requires a very different approach for rubber than for steel. It would probably be desirable to identify rubber compounds by their modulus in the required range, as this is a much more accurate and useful index than the durometer reading, taken at low distortion. To store equal energy in two pieces of rubber, it is necessary that the products of stress times strain be equal. It is; therefore, desirable that for similar nominal static deflections, the static stress should vary as the square root of the modulus or:

$$S_2 = S_1 X \sqrt{\frac{G_2}{G_1}}$$

Creep in rubber springs is becoming a relatively unimportant factor. There are several well proved methods of presetting rubber springs to take out the more rapid early creep. While it is true that some small creep may continue through the life of a rubber spring, this should not be of objectionable magnitude for a well-designed rubber spring. We have operated a rubber sprung car for 2½ years without adjustment, without benefit of presetting.

Mr. Schilling says that low rubber stresses may be used to avoid excessive creep. In general, creep should be measured in terms of per cent of static deflection. In these terms many good compounds show very little change in creep through a wide stress range, particularly at the range of temperatures to which suspension springs are exposed.

The greatest amount of progress can be made only by a realistic evaluation of all spring materials and all types of springs. We fall easily into the error of minimizing the short comings of existing springs because we are accustomed to them. It would not be good engineering to ignore the advantages which led us to select some recognized type of spring in the first place. However, with years of use each type of spring has progressed, and in selecting a new type of spring, we should look both at its present merits and also at the possibility that, if it is now competitive in a technical sense, it may far outstrip the older types of springs after its wider use has attracted the thought and ingenuity of a larger number of engineers.

■

Increased Static Deflection Held Watchword of Designers

—BERNHARD STERNE
Chrysler Corp.

THE author has made it clear that the choice of spring cannot be based properly on any single consideration. Even the basic criterion of energy storage per unit weight of material does not always furnish an unequivocal answer when the weight of the entire mechanism is taken into consideration.

The leaf spring, which always makes an extremely poor showing in such comparisons, has so far retained an important place in vehicle design because of its low cost and its versatility in doubling as a structural member. Some good ride engineers welcome a limited amount of static friction as long as it does not fluctuate too much.

Not nearly enough of the methods have been explored so far to obtain the load-deflection curve with variable rate which Mr. Schilling has discussed. The future mastery of the art to combine rate controlling influences of linkage and of bumper springs should restore the suspension spring to its main task of furnishing ample deflection.

The exercise of some ingenuity is also called for to find methods

of combining this ample spring deflection with a reduced change in standing height of the vehicle.

Many a truck engineer will then be glad to avail himself of the economy in spring design which results from increasing the static deflection to the equivalent of the ride clearance, with benefits to the driver, the payload, and the life of the truck.

Increased static deflection has become the watchword not only of the ride engineer but of the spring engineer himself. True, the limitations in maximum stress impose the demand for proportional weight increase with each increase in deflection. And yet, there is a compensating feature, as a reduction in load rate means a reduction in stress rate, which is equivalent to a reduction in stress range for a given amplitude and therefore to an increase in fatigue life.

The most obvious benefit to the spring life occurs when the static deflection is made large enough to prevent rebound past the zero load position and thus loss of residual stresses which were trapped by presetting and possibly by shot peening.

The significance of these two operations has emerged more and more clearly during the last 10 years. Yet we cannot claim to have understood fully the underlying principles or to have gained sure knowledge of the most efficient techniques. Research along these and other lines will go on for many a year to come in our continued quest for the best in suspension springs.

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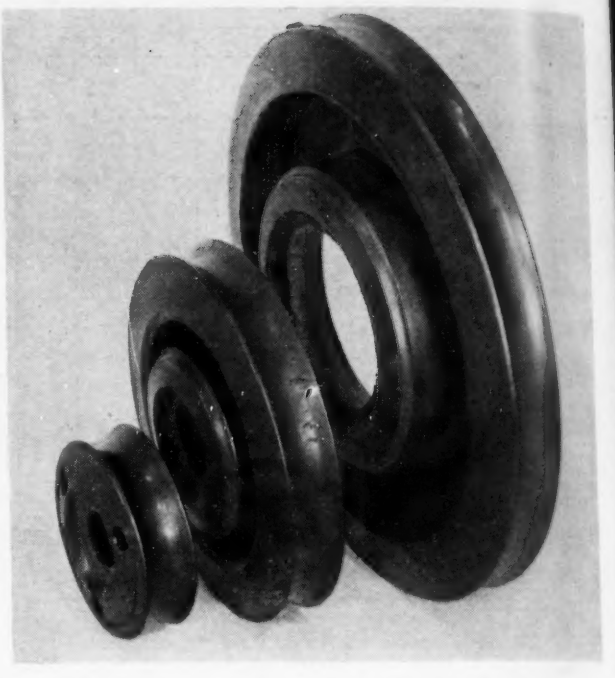

■ Fig. D – Low weight rubber spring discs – maximum load capacities for sizes shown are 14,000, 30,000 and 100,000 lb – energy storage capacity approximates 5000 in.-lb per lb of spring

Says Rubber Promises Lower Spring Weight

– ROY W. BROWN

Firestone Tire & Rubber Co.

IT has been shown that the best steel spring can store 1000-1500 in.-lb of energy per lb of spring. Rubber in shear will store 2000-4000 – double that of steel.

A conical disc construction, shown in Fig. D, using rubber in compression was developed for aircraft undercarriages[a]. These discs provide for a maximum energy storage of 5000 in.-lb per lb of rubber spring at 50% deflection.

The rubber bellows enclosure for air springs offers still greater reduction in spring weight. Thus, the suspension designer has a choice of promising materials to work with in the event he desires to forget established assumptions and engineer a completely new suspension.

■ Dynamic Characteristics

The motor-car user has little interest in spring characteristics when the car is standing still. It is only under moving or dynamic conditions that spring performance becomes objectionable or otherwise. Hence, it becomes necessary to study, measure, and evaluate the very complex vibration movement which occurs when the car is propelled over an irregular road surface.

It is generally assumed that such vibration movement approximates simple harmonic motion. A glance at an accurate sensitive record of a good car on a good road indicates that movement is very complex and occasionally far removed from harmonic motion.

An analysis can best be made by separating each movement into a series of smaller movements and then determining for each, frequency, acceleration, and the like on the basis of assumed harmonic motion. Another method is to consider each movement as a transient and determine steepness of wave front.

In any event a more rigorous analysis of motor-car vibration movement shows a large number of important higher frequency vibrations which may be classed as secondary vibration. It is important to evaluate the spring medium from this point of view.

■ Secondary Vibration

It has been shown that any condition which causes the dynamic spring rate to change too quickly is an undesirable cause of secondary vibration. Such rate change may result from interleaf friction, shackles, lost motion, short sway bar links, and shock absorber action. Hence, the spring medium enabling the most direct attachment is favorable to reduction of the unwanted secondary vibrations.

Of all experimental suspensions now proposed the air spring[b] is the only one which promises a spring medium combined with the shock absorber and suitable for direct attachment. It remains to determine detail construction suitable for the varied conditions encountered under present-day service.

The location of the spring medium between the axle and the body provides a path from the road to the passenger for transmission of the undesirable higher frequency vibrations. Resistance to such transmission is, therefore, important. It varies widely for different materials. A numerical comparison made to air as 1.0 shows rubber 0.002 and steel 0.00001. Thus, the resistance to transmission of high-frequency vibration through air and rubber is enormously greater than through steel.

■ Why Not Aircraft Technique?

A recent English technical magazine[c] editorial emphasized the intriguing characteristics of aircraft undercarriages for motor-car suspensions. Previously stated values of weight and energy storage are dwarfed when it is considered that a new type undercarriage weighs only 31 lb (including tire, wheel, brake, spring medium, shock absorber, and attachment) and carries a static load of 1000 lb with a wheel travel of 10 in. and can handle 32,000 in.-lb of energy. Similar motor-car parts will weigh 4-5 times as much.

Another spring medium, liquid springs, has been introduced by the aircraft industry[d]. Compressibility of liquids and extensibility of the containing cylinders enables low displacement at high pressures to provide a flexible load support. Sealing problems and extreme precision of construction appear to limit present application to special devices.

The experienced suspension engineer will rightly note many differences in requirements. However, it may well be that visionary thinking not yet too well acquainted with established assumptions, will prevail and a practical low-weight composite high-performance motor-car suspension may result from aircraft undercarriage practice.

[a] See SAE Journal (Transactions), Vol. 53, March, 1945, pp. 177-183: "Research and Tomorrow's Aircraft Undercarriages," by Roy W. Brown.

[b] See SAE Journal (Transactions), Vol. 38, April, 1936, pp. 126-132: "Air Springs – Tomorrow's Ride." by Roy W. Brown.

[c] See *Automobile Engineer*, Vol. 35, July, 1945, p. 257.

[d] See *Aeroplane*, Vol. 66, Feb. 25, 1944, p. 221: "Undercarriage Development"

THE ROCKET POWERPLANT[1]

by M. J. ZUCROW
Aerojet Engineering Corp.

THE jet propulsion action of the rocket has been recognized for centuries, but its first use cannot be stated with certainty. An old Chinese manuscript described their use by the Chinese against their enemies in the battle of Pien-king, 1232 A.D. The rockets were attached to conventional arrows to increase their range. It seems probable, however, that the Greeks recognized the rocket principle several centuries earlier.[2, 3] For details of the early history of the rocket refer to Ley,[2] Zim,[3] and Goddard.[4]

Since its earliest beginnings the application of the rocket has oscillated between its use in wartime as a weapon and its use in peace for signaling or pyrotechnic displays. Its employment as a weapon was practically eliminated during the latter part of the nineteenth century by the advances made in artillery, which had greater range and accuracy. In World War I, the rocket was employed only to an insignificant extent. In World War II, however, it reappeared as a major offensive weapon employed by all of the warring powers, and also as a means for propelling aircraft and missiles.

The basic developments, both theoretical and experimental, which brought the rocket to its present prominence must be credited to the imagination and research of such pioneers as Ziolkovsky and Rynin in Russia; Oberth, Valier, Hohmann, and Sänger in Germany; Esnault-Pelterie in France; and the late Dr. R. H. Goddard in the United States. In this connection the work of Dr. Goddard was preeminent, for he conducted the first scientific experiments and published the first noteworthy treatment of this subject. Other scientific works quickly followed.[5, 6]

The pioneers mentioned above analyzed the rocket and saw that it had potentialities as a means of attaining extreme altitudes and speeds, and for making space travel a possibility. Despite many discouragements and even ridicule they persisted in their efforts to demonstrate to the scientific world that the rocket principle possessed the potentialities for attaining the above objectives.

The particular advantages of the rocket that are possessed by no other propulsion means known at this time are:

1. Its thrust is almost independent of its environment.
2. It requires no atmospheric oxygen for its operation.
3. It can function best in a vacuum.
4. It is the simplest known means for converting the thermochemical energy of a propellant combination (fuel plus oxidizer) into the kinetic energy associated with a jet of fast moving gases.

Rocket jet propulsion should not be regarded as a competitor of existing means for propelling aircraft, but as a source of power for attaining objectives unattainable by other methods. The Germans applied rocket propulsion to the high-speed, short-duration ME-163 interceptor, and to the V-2 missile. In this country, apart from its application to weapons, the application of rocket jet propulsion to aircraft has been concerned mainly with its use as an auxiliary to the propeller for assisting in the take-off.

The purpose of this paper is to present certain information regarding the fundamentals of rocket jet propulsion and its application to the assisted take-off of aircraft, being limited to information on the *constant-pressure rocket.*

The propelling action of a rocket motor is derived from the generation of large quantities of high-temperature gases from the chemical reaction of suitable propellants.

Fig. 1 illustrates the principal elements of a bipropellant rocket motor utilizing a liquid oxidizer and a liquid fuel; each liquid is called a propellant. The rocket motor comprises a combustion chamber, an exhaust nozzle, a propellant injection system, and remotely operated propellant control valves. No ignition system is shown, because propellants can be selected that react spontaneously on impingement of their liquid streams. The propellant gases are produced in the combustion chamber at a pressure governed by the characteristics of the propellants employed, the rate of propellant consumption, the mixture ratio (oxidizer weight flow rate/fuel weight flow rate), and the area of the nozzle throat section. The propellant gases are ejected into the atmosphere with supersonic velocity. The combustion-chamber pressure p_c ranges from 300 to 750 psi (absolute), depending upon the characteristics of the propellants.

a. Liquid Bipropellant Rocket Jet Propulsion System – Fig. 2 illustrates the essential components of a complete liquid bipropellant rocket jet propulsion system. The system consists of the oxidizer and fuel tanks, means for pressurizing the propellants to force them through the injection system against the chamber pressure, propellant control valves, and the rocket motor. The differential pressure across the injection system must be sufficiently high to give satisfactory atomization of the propellants.

The liquid rocket jet propulsion system is capable of operation for extended periods, limited only by the installed capacity of the propellant supply tanks.

For short duration applications the pressurizing of the propellants can be accomplished by incorporating a high-

[This paper was presented at a meeting of the Southern California Section of the SAE, Los Angeles, Calif., Jan. 11, 1946.]

[1] Based on part of Chapter II of "Principles of Jet Propulsion," by the author. To be published by John Wiley & Sons, Inc., New York, 1946.

[2] See "Rockets," by W. Ley. Published by Viking Press, New York, 1944.

[3] See "Rockets and Jets," by H. S. Zim. Published by Harcourt, Brace, and Co., New York, 1945.

[4] See Smithsonian Institute Misc. Coll., Vol. 71, No. 2, 1919, "Method of Reaching Extreme Altitudes," by R. H. Goddard.

[5] See "Die Rakete zu den Planetenräumen," by H. Oberth. Published by R. Oldenbourg, Munich, Germany, 1925.

[6] See "Raketenflugtechnik," by E. Sänger. Published by R. Oldenbourg, Munich, Germany, 1933.

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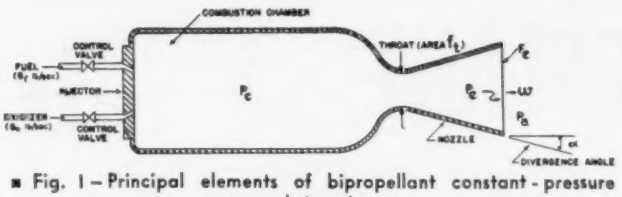


■ Fig. 1 – Principal elements of bipropellant constant-pressure rocket motor

pressure inert gas supply tank, as illustrated in Fig. 3.

For longer durations a turbine-driven, pumping system, similar to that used in the German V-2 missile may be employed.[7] These utilize the chemical reaction of propellants to furnish the gases for driving the turbine. The propellants need not necessarily be the same ones that are used for operating the rocket powerplant. Liquid systems can be started and stopped at will by the remote operation of the requisite control valves.

b. Solid Propellant Rocket Motors – Fig. 4 illustrates a rocket motor utilizing a solid propellant: fuel plus oxidizer. This type of rocket motor has been used most extensively during the war for the assisted take-off of aircraft. To obtain the desired action the characteristics of the propellant charge must be so designed that it burns at a uniform rate (at substantially constant pressure) in layers perpendicular to the longitudinal axis of the motor; that is, the burning is restricted to take place in one direction only. One method for accomplishing this is by bonding the propellant charge to the steel wall of the motor with a special flexible material or liner which allows the propellant the freedom of motion required for adapting its shape to the deformations due to gas pressure, thermal expansion, and internal stresses. An electrically operated igniter is used for initiating the combustion of the propellant. The operating chamber pressure ranges from approximately 1200 to 2800 psi (absolute).

Fig. 5 is a photograph of the exterior of a motor which was built so that it can be loaded with a propellant charge which is in the form of a reloadable cartridge. The photograph shows the exterior of the motor chamber and the detachable forecap, which is removed to insert the cartridge. Fig. 6 is a cut-a-way section through the loaded motor. It shows the cartridge type of propellant charge, the igniter for initiating combustion, the safety pressure release diaphragm assembly, the attachment lugs, and the detachable forecap.

Rocket motors employing a restricted burning solid propellant are used for applications where a constant thrust of relatively short duration, up to 45 sec, is useful. In addition to their application to the assisted take-off of aircraft, they can be applied to the propelling of projectiles and missiles, and for signaling devices. They can also be applied to those special services requiring a relatively limited duration of high-temperature gas for power generation purposes. Solid propellant rocket motors have been built that can be stopped at will, but they cannot be restarted, as may be done with liquid units.

Refer to Fig. 1 and let it be assumed that the chamber pressure is constant and the flow of exhaust gases through the nozzle is steady. The thrust developed by the rocket motor under these conditions is given by:[8]

$$T = (p_e - p_a) F_e + \frac{G}{g} \lambda c_e \qquad (1)$$

The factor λ can be calculated for any divergence angle of the nozzle exit cone from the equation:

$$\lambda = \frac{1}{2} + \frac{1}{2} \cos \alpha \qquad (2)$$

Equation (1) indicates the manner in which the atmospheric pressure surrounding the rocket motor affects its thrust. The lower the external pressure the larger is the thrust, and the maximum thrust is attained when $p_a = 0$. In other words, *the rocket motor develops its maximum thrust when operated in a vacuum.*

Fig. 7 presents the factor λ as a function of the divergence angle α. Since the divergence angle of most rocket motor nozzles is less than 15 deg, it can be assumed for practical purposes that $\lambda = 1$.

Consider a rocket motor operating with constant chamber pressure and let the exit area F_e be gradually increased. It is assumed that the increase in exit area is obtained with the same nozzle divergence by gradually increasing the length of the exit cone. The nozzle is designed to operate with the critical pressure in the throat section so that the velocity in the throat is the local sonic velocity, in the exit cone the velocities are supersonic. Consequently, conditions in throat are undisturbed by increasing exit area.

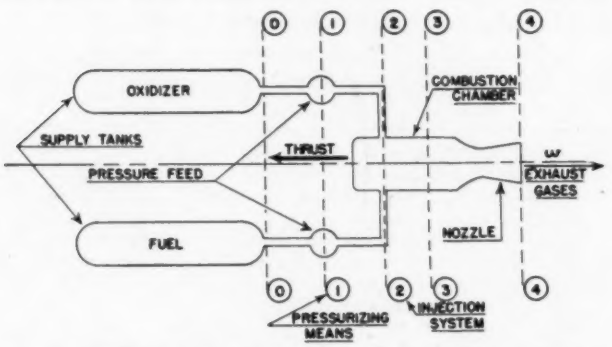
The latter follows from the nature of compressible flow at supersonic velocities.[9] For when the velocity of a gas is subsonic a pressure wave which can move only with the sonic velocity can propagate itself against the direction of the flow. Consequently, any restriction or obstruction in the flow passage causes the gas flowing into it to adjust itself to suit the conditions. When the flow is supersonic a pressure wave due to a disturbance, since it travels at the sonic speed, cannot propagate itself upstream against the higher velocity of the inflowing gases.[10]

As the exit area is increased the exhaust velocity of the gases crossing it is also increased, because of their greater expansion, in accordance with the equation:

$$\frac{c_e}{c_t} = \sqrt{\frac{k+1}{k-1}\left[1 - \frac{2}{k+1}\left(\frac{p_e}{p_t}\right)^{\frac{k-1}{k}}\right]} \qquad (3)$$

The ratio p_e/p_t, of the pressure at the exit section to the throat pressure, is related to the area F_e/f_t by:

[7] See *Life*, Vol. 17, Dec. 25, 1944, pp. 46-48: "V-2 Nazi Rocket Details Are Finally Revealed."
[8] See Journal of Franklin Institute, Vol. 230, October, 1940, pp. 433-454: "Characteristics of Rocket Motor Unit Based on Theory of Perfect Gases," by F. J. Malina.
[9] See ASME Transactions (Journal of Applied Mechanics), Vol. 11, June, 1944, pp. A-93-A-100: "Nozzles for Supersonic Flow without Shock Fronts," by A. H. Shapiro.
[10] See Journal of Aeronautical Sciences, Vol. 11, July, 1944, pp. 227-238: "Thermodynamics of Air at High Velocities," by N. P. Bailey.



■ Fig. 2 – Elements of bipropellant liquid rocket jet propulsion system

$$\frac{F_e}{f_t} = \left(\frac{p_t}{p_e}\right)^{\frac{1}{k}} \frac{c_t}{c_e} \qquad (4)$$

Fig. 8 presents plots of the ratio of velocities at the exit and throat sections c_e/c_t and the expansion ratio p_e/p_t as functions of the area ratio F_e/f_t. It is apparent from this figure that as the exit area is first increased the exit pressure decreases at a much more rapid rate. Consequently, the product p_eF_e diminishes while the product p_aF_e increases. The net result is a reduction in the magnitude of the first term $(p_e - p_a)\ F_e$ in the equation for the rocket motor thrust. The exit velocity c_e is, however, greatly increased by the increased expansion so that the net result is an increase in thrust. The thrust continues to increase as the area ratio F_e/f_t is made larger until the exit pressure p_e is equal to that of the surrounding atmosphere. At this area ratio value the term $(p_e - p_a)F_e = 0$. The gases are completely expanded to the atmospheric back pressure and attain their full velocity corresponding to an expansion from the combustion-chamber pressure to the atmospheric pressure. Before this condition is attained a portion of the useful energy of the propellant gases is not converted into kinetic energy, so that it is lost so far as thrust development is concerned. When the nozzle functions under the condition where the exit pressure p_e is greater than the back pressure p_a, it is said to operate with *underexpansion*.

If the area ratio is increased beyond the value required to make $p_e = p_a$, then the exit pressure of the gases drops below that of the atmospheric back pressure. $(p_e - p_a)F_e$ becomes negative and thrust is reduced. When this occurs the nozzle is said to operate with *overexpansion*.

When operated with overexpansion or if the nozzle divergence angle is too large the pressure at some section in the exit cone may be too low to maintain itself against the back pressure and there occurs a sudden increase in pressure at that section. Beyond that section the velocity falls to a subsonic value. This discontinuous rise in pressure is called a shock wave. The shock wave front is very thin, of the order of a few millionths of an inch.[11] The sudden drop in the exhaust velocity due to the shock wave destroys the value of the nozzle. The effect of the shock on the velocity and pressure distribution beyond it is illustrated schematically in Fig. 9.

It should be noted that when operated with overexpansion a shock wave may occur in the nozzle exit cone, but before the overexpanded condition is reached, the jet will probably separate from the walls of the nozzle.[8] Because of these phenomena, most rocket motor nozzles are designed to operate with a slight underexpansion, so that supersonic velocities are always maintained in all sections of the exit cone.

The area ratio for complete expansion can be obtained from the following equation:

$$\frac{f_t}{F_e} = \left(\frac{k+1}{2}\right)^{\frac{1}{k-1}} \left(\frac{p_a}{p_e}\right)^{\frac{1}{k}} \sqrt{\frac{k+1}{k-1}\left[1 - \left(\frac{p_a}{p_e}\right)^{\frac{k-1}{k}}\right]} \qquad (5)$$

Fig. 10 presents the area ratio for complete expansion as a function of the pressure ratio for several values of k.

The corresponding velocity ratio can be expressed in terms of the expansion ratio p_a/p_e by substituting for p_t in equation (3), and assuming full expansion to the atmospheric back pressure p_a, so that $p_e = p_a$. Thus:

$$p_t = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} p_e$$

$$\frac{c_e}{c_t} = \sqrt{\frac{k+1}{k-1}\left[1 - \left(\frac{p_a}{p_e}\right)^{\frac{k-1}{k}}\right]} \qquad (6)$$

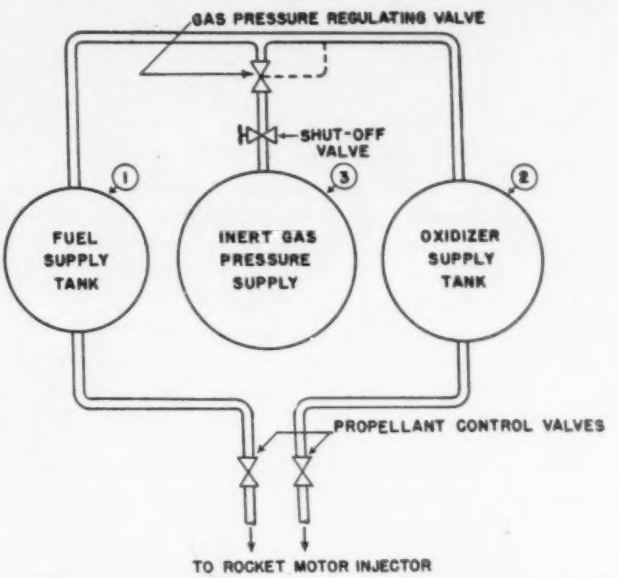
$$c_t = \sqrt{2g\frac{k}{k+1}RT_e}$$

The value of the gas constant R will depend upon the composition of the propellant gases, combustion-chamber pressure, and temperature.

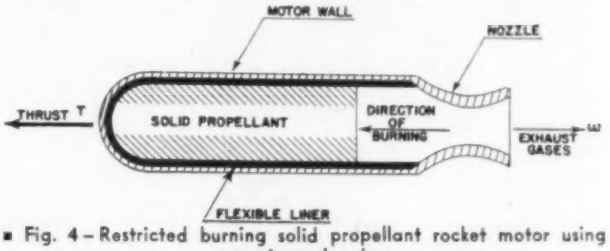
Since the atmospheric back pressure affects the thrust of the rocket motor, this must be given consideration if the rocket motor is to operate through a range of altitudes. To avoid jet separation and shock at low altitudes where the back pressure is high, it is customary to design the nozzles so that it operates with a slight underexpansion at sea level.

Fig. 11 is the so-called thrust diagram developed by Dr. F. J. Malina[8] for the convenient determination of the thrust variation for a given rocket motor at different operating altitudes; that is, with varying back pressure. The ordinate of this chart is the ratio of the total thrust developed by the rocket motor T to the thrust based on the nozzle throat area T_t. The abscissa is the area ratio F_e/f_t.

The total thrust T (lb) is determined from the equation:



■ Fig. 3 – Elements of inert gas pressurizing system for liquid rocket jet propulsion system



■ Fig. 4 – Restricted burning solid propellant rocket motor using one-piece chamber

[11] See Vol. I, "Steam and Gas Turbines," by A. Stodola. Published by McGraw Hill Book Co., Inc., New York, 1927 (out of print).

$$T_t = \left[2 \left(\frac{2}{k+1} \right)^{\frac{1}{k-1}} p_c - p_a \right] f_t \qquad (7)$$

and the thrust ratio T/T_t is given by the diagram.

Fig. 12 presents the thrust-altitude characteristics of two rocket motors having different area ratios.

The thrust of the rocket motor, as given by equation (1), can be expressed in a form that is more convenient for studying its performance. Thus, let:

$$T = \frac{G}{g} w = (p_e - p_a) F_e + \frac{G}{g} \lambda c_e \qquad (8)$$

Assuming that $\lambda = 1$, then w, which is called the *effective exhaust* velocity, in fps, is given by:

$$w = c_e + (p_e - p_a) F_e \frac{g}{G} \qquad (9)$$

The effective exhaust velocity is the basic criterion of rocket motor performance. It is apparent that the effective exhaust velocity is larger than the exit or muzzle velocity of the propellant gases c_e, and its value depends entirely upon the conditions within the rocket motor. If the area ratio F_e/f_t is such that the exit pressure is equal to the back pressure, so that $p_e = p_a$, then with $\lambda = 1$ the effective exhaust velocity is identical with the exit velocity.

In the static test of a rocket motor the measured variables are the thrust, total weight of propellants consumed Gp, and the duration of the run t. From these data the effective exhaust velocity is calculated. Thus:

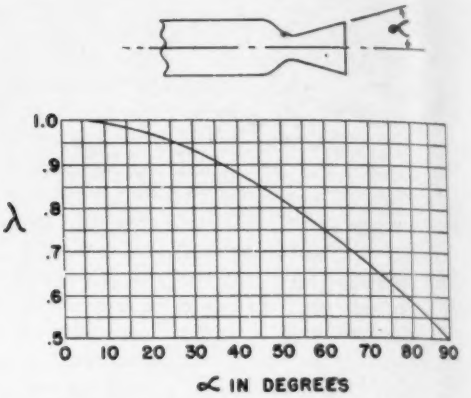
$$w = T \frac{g}{G} = Tg \frac{t}{G_p} \qquad (10)$$

The propellant consumption rate of a rocket motor can be readily determined from its effective exhaust velocity. Thus it has been reported that the effective exhaust velocity of the German V-2 missile is 6400 fps, and its thrust is 48,000 lb. The rate of propellant consumption is found from equation (10) to be:

$$G = \frac{48{,}000 \times 32.174}{6400} = 241 \text{ lb per sec}$$

■ Fig. 5 (left) – Restricted burning solid propellant rocket motor using two-piece chamber (cartridge type of unit)

■ Fig. 6 (right) – Cartridge type of solid propellant unit



■ Fig. 7 – Nozzle divergence correction factor λ as function of divergence angle α

For the steady state conditions assumed here, it is possible to calculate the effective exhaust velocity from the thermodynamic properties of the propellant gases and their expansion ratio. Thus:

$$w = \phi \sqrt{2g \frac{k}{k-1} RT_c \left[1 - \left(\frac{p_e}{p_c} \right)^{\frac{k-1}{k}} \right]} \qquad (11)$$

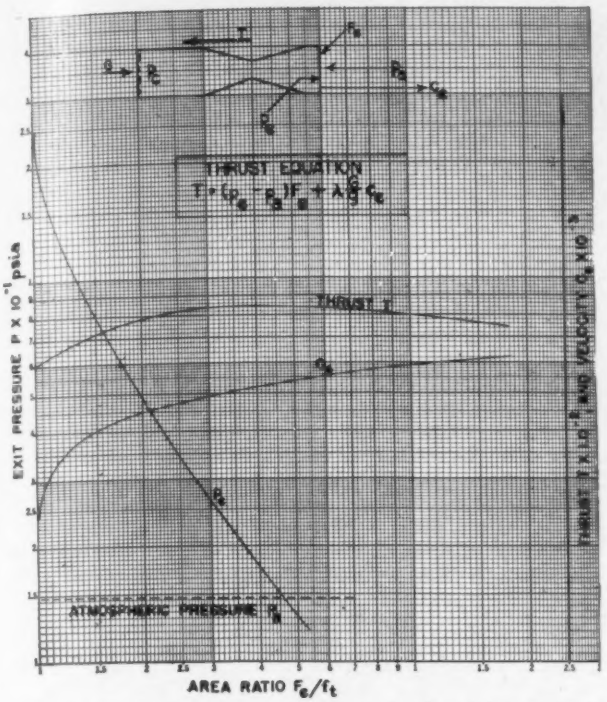
The above equation shows that the effective exhaust velocity increases directly as the square root of the product of the combustion-chamber temperature T_c and the gas constant R. It also increases with the expansion ratio and with a decrease in the specific heat ratio for the gases. The two last mentioned factors exert only a minor influence, so that for practical purposes it can be assumed that the effective exhaust velocity is proportional to $\sqrt{T_c R}$. Since the gas constant $R = 1545/m$, where m is the molecular weight of the gases, it follows directly that the effective exhaust velocity is proportional to $\sqrt{T_c/m}$.

Fig. 13 illustrates the manner in which the combustion-chamber temperature and pressure affect the effective exhaust velocity, assuming that the molecular weight is constant and $k = 1.2$.

For the gases generated by the propellants currently being used R can be assumed to vary between 60 and 80.

The thrust delivered per unit weight rate of propellant consumption, termed the *specific impulse*, is frequently used as one of the principal criteria of rocket motor performance. The specific impulse (lb per lb per sec) is defined by:

$$I_s = \frac{T}{G} = \frac{w}{g} \qquad (12a)$$

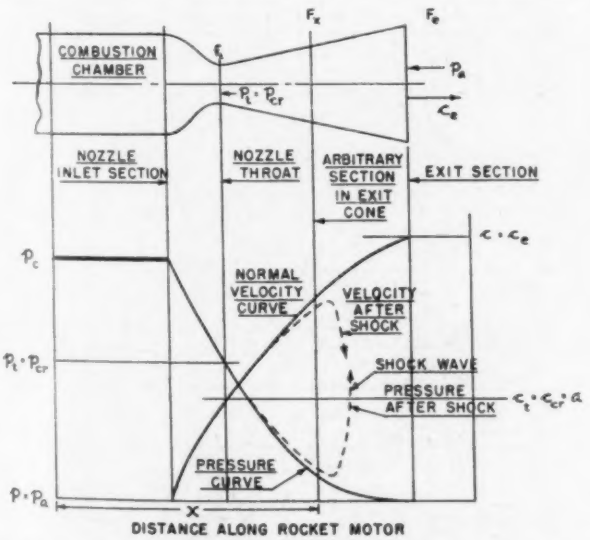


■ Fig. 8 – Effect of area ratio on thrust developed by constant-pressure rocket motor

Substituting for the effective exhaust velocity from equation (11) and noting that $R = 1545/m$, where m is the molecular weight of the propellant gases, then:

$$I = \frac{w}{g} = 692\,\phi\,\sqrt{\frac{T_c}{m}}\,\sqrt{\frac{2k}{k-1}\left[1-\left(\frac{p_e}{p_c}\right)^{\frac{k-1}{k}}\right]} \qquad (12b)$$

This last equation shows that the specific impulse depends primarily upon the square root of the ratio of the temperature of the gases in the combustion chamber to their molecular weight; this follows from the preceding discussion of the equation for the effective exhaust velocity.



■ Fig. 9 – Effect of shock wave on pressure and velocity distributions in exit cone

The maximum temperature in the combustion chamber is limited by dissociation reactions and, from a practical standpoint, by the materials available for constructing rocket motors. Consequently, the possibility of increasing the specific impulse by raising the combustion-chamber temperature is limited.

The possibility of increasing the specific impulse by reducing the molecular weight of the gases by using fuels rich in hydrogen also suffers from certain restrictions. Fuels rich in hydrogen have low specific weight, so that the average specific weight for the propellants is reduced. This makes it necessary to use supply tanks having large volumes, thereby greatly increasing the overall weight of the jet propulsion system. For this reason a more useful practical guide to the suitability of propellants is the product (specific weight × specific impulse).

The adequacy of a rocket motor depends almost entirely upon its ability to perform without damage at the high temperatures for the operating duration, and the number of runs, to which it is subjected. An appreciation of the magnitude of the heat problem is obtained if it is realized that the heat release per unit volume of the combustion chamber is 100 to 150 times that encountered in modern furnaces for high-pressure high-temperature steam generation. Further, depending upon the propellants, this heat is generated at from 4000 to 6000 deg Rankine.

Of the total heat liberated approximately 5% is transferred to the motor and nozzle walls; this amounts to 120 to 200 Btu per sec. The transferred heat can be handled by three methods: (1) the motor walls and nozzle can be constructed of suitable materials and heavy enough to absorb the heat during the operating period; (2) one of the propellants in its passage to the injection system can be circulated around the heat absorbing surfaces to keep them cool – this is the regenerative cooling system and decreases the heat losses; and (3) high-temperature-resistant (refractories) materials can be used for lining the heated surfaces.

Method (1) employs the principle of supplying a heat reservoir capable of receiving the total quantity of heat to be absorbed without raising the metal temperatures to dangerous values. The most suitable materials for this type of cooling are those for which the product (specific heat × thermal conductivity × density) has high values. This can be verified by dimensional analysis or by experiment; the best material from this standpoint is copper. The foregoing criterion is not, however, a unique guide to motor construction, since considerations of strength and weight are frequently of greater importance. In any case, as the required operating duration for an uncooled motor is raised, the requisite motor weight becomes excessive for practical use. Consequently, this type of motor construction is adaptable only to rocket motors which are to operate for short durations and may be operated with high fuel oxidizer ratios to reduce the operating temperature. Satisfactory motors of this type have been built for durations up to 35 sec. They usually weigh more than rocket motors of equal thrust output that employ regenerative cooling.

Method (2), regenerative cooling, appears currently to be the best approach to the solution of the heat problem. It has the advantage that once the cooling system has been developed correctly, the motor can be operated for long durations (several minutes at a time) without damage.

Furthermore, these motors can be made very light, thrust-weight ratio increasing markedly with larger thrusts.

The principle of regenerative cooling is illustrated in Fig. 14 which represents a small rocket motor using liquid oxygen and gasoline tested by E. Sänger in 1932.

The little data that have been published regarding method (3) are encouraging. It seems probable that by combining methods (2) and (3) higher operating temperatures may become feasible.

The thrust developed by a rocket motor is the resultant of the combustion gas and external pressures acting in the direction of the axis of the motor (see Fig. 1). Under the steady state conditions assumed here and for $\lambda = 1$, the thrust (lb) according to equation (10) is given by:

$$T = \frac{G}{g} w \tag{13}$$

It can be shown by thermodynamic reasoning that the thrust is also given by the equation:

$$T = C_T p_c f_t \tag{14}$$

where C_T is called the *thrust coefficient* and is defined by:

$$C_T = \psi \sqrt{\left(\frac{k^2}{k^2-1}\right)\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}\left[1-\left(\frac{p_e}{p_c}\right)^{\frac{k-1}{k}}\right]} \tag{15}$$

The value of the coefficient C_T will depend upon the motor design and the specific propellants. The value of ψ will range from 0.85 to 0.96, a good average value being 0.936 for propellant gases for which k may be assumed to have the value $k = 1.2$.

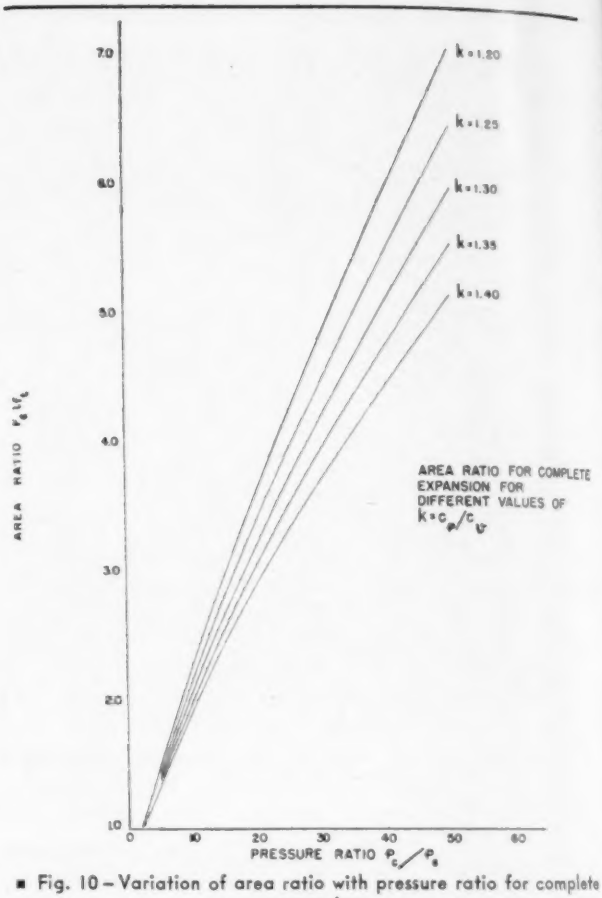
Fig. 15 presents the values of C_T as a function of combustion-chamber pressure, assuming $\psi = 1.0$, and $\psi = 0.936$.

Equation (14) is one of the basic design formulas for rocket motors. The value of C_T as a function of combustion chamber is determined experimentally for the particular propellants to be employed. This is readily accomplished since the thrust, combustion-chamber pressure, and throat area can be measured accurately. The curve presenting C_T as a function of chamber pressure gives data for establishing the basic dimension of the nozzle for rocket motors of any desired thrust development or operating chamber pressure.

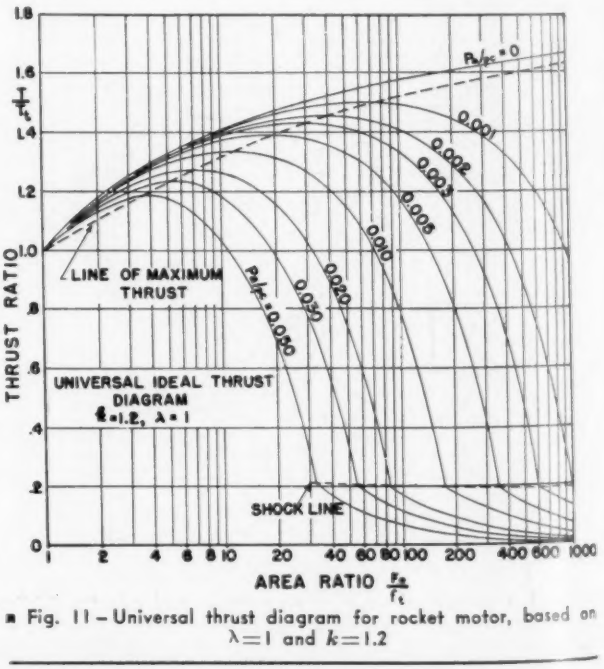
The energy utilized by the rocket motor for propulsion purposes is derived from the thermochemical energy of the propellants. Since the fuels employed are predominantly hydrocarbons the combustion products contain CO_2, CO, and H_2O as principal constituents. Because of the high temperatures attained these gaseous products are dissociated. Thus, the CO_2 formed dissociates into $CO_2 = CO + \frac{1}{2} O_2$, the water vapor into $H_2O = H_2 + \frac{1}{2} O_2$. If the temperature is sufficiently high there may occur a breakdown of the above molecular products. For example, at 4500 F the water vapor dissociates into $H_2O = H + OH$, at 7200 F it dissociates into $H_2O = H + H + O$, and at 9000 F no molecular components are present.[12] The degree of dissociation increases with the temperature and decreases with the pressure.[13]

The net result of the dissociation phenomena is that the maximum temperatures reached in the combustion chamber are much lower than those calculated upon the basis of the complete combustion of the fuel and oxidizer.

The final effect is that a portion of the calorific value of the fuel is unavailable for transformation into kinetic energy for propulsion. From a practical standpoint this is not serious, because any phenomenon which reduces the



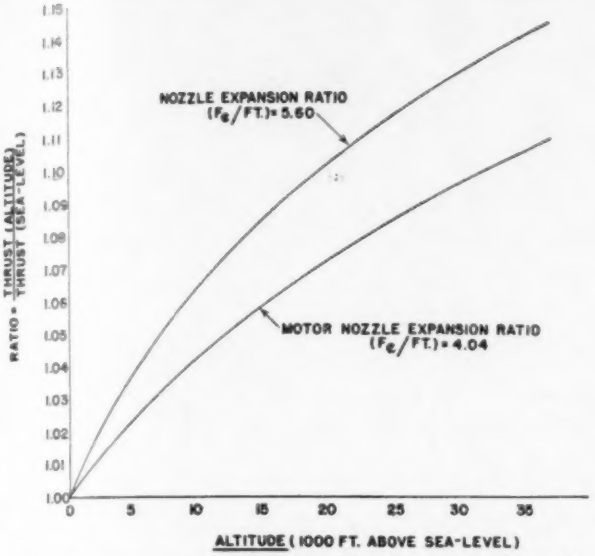
■ Fig. 10 – Variation of area ratio with pressure ratio for complete expansion



■ Fig. 11 – Universal thrust diagram for rocket motor, based on $\lambda = 1$ and $k = 1.2$

[12] See Journal of American Chemical Society, Vol. 37, 1915, pp. 417-458: "Dissociation of Hydrogen into Atoms, Part II," by I. Langmuir.

[13] See *Automobile Engineer*, Vol. 11, February, March, April, 1921, pp. 55-59, 98-101, 134-137: "Character of Various Fuels for Internal-Combustion Engines," by H. T. Tizard and D. R. Pye.



■ Fig. 12 – Effect of atmospheric back pressure upon thrust developed for two rocket motors having area ratios of 4.04 and 5.60

maximum flame temperature is beneficial, due to the limitations imposed by the materials available.

In addition to the energy loss due to dissociation, there are losses due to radiation, fluid friction, pumping work, imperfect combustion, and the like.

The effect of combustion-chamber pressure upon the propellant gas composition depends upon the subsequent reactions which can take place between the products of combustion. When the possible reaction between any of the products is such that the volume of these reactants is larger than that obtainable from their complete chemical reaction, increasing the pressure favors driving the chemical reaction to completion. If the completed chemical reaction does not produce a change in volume, then pressure has no effect. On the other hand, if the completed chemical reaction will give an increase in volume, then increasing the pressure hinders the completion of the reaction. This is summarized in Table 1.

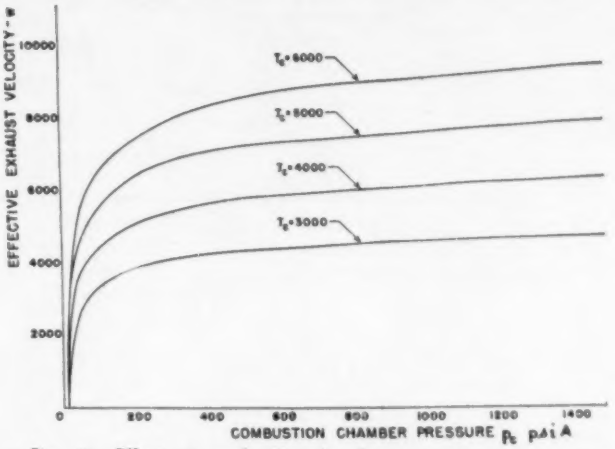
Table 1 – Effect of Pressure on Completeness of Reaction

Reaction Equation ($_g$ = gas; $_s$ = solid)	Volume of Reactants, mol	Volume of Products, mol	Effect of Pressure on Completion
$(H_2)_g + \frac{1}{2}(O_2)_g = (H_2O)_g$	1.5	1.0	Favors
$(CO)_g + (H_2O)_g = (CO_2)_g + (H_2)_g$	2.0	2.0	No effect
$(C)_s + \frac{1}{2}(O_2)_g = (CO)_g$	0.5	1.0	Hinders

Consider a rocket jet propelled system moving with the constant flight speed V. Assume that the rocket motor operates with a constant chamber pressure p_c, so that the rate of propellant consumption G and the effective exhaust velocity w are constants. The thermochemical energy of the propellants is converted into kinetic energy for propulsion purposes. A part of this energy is utilized to impart kinetic energy to the unconsumed propellants moving at the flight speed V. A part is used to produce useful work or the thrust power P_T, and the balance is lost with the ejected gases as unconsumed kinetic energy; the latter is called the leaving loss P_L.[14]

The total work made available for propulsion purposes at any instant is at the rate (ft-lb per sec):

[14] See ASME Transactions, Vol. 68, April, 1946, pp. 177-188: "Jet Propulsion and Rockets for Assisted Take-Off," by M. J. Zucrow.



■ Fig. 13 – Effect of combustion-chamber pressure and temperature upon effective exhaust velocity

$$P = \frac{G}{2g} w^2 + \frac{G}{2g} V^2 = \frac{G}{2g} w^2 (1 + v^2) \qquad (16)$$

where P is termed the propulsive power, and $v = V/W$ is termed the velocity ratio.

The thrust power (ft-lb per sec) developed by the rocket motor is given by:

$$P_T = TV = \frac{G}{g} wV = \frac{G}{g} w^2 v \qquad (17)$$

It is of interest to compare the above equation for the thrust power of the rocket with that for the *propeller* and for the thermal jet engine. For the latter two systems:

$$P_T = TV = \frac{G}{g} w^2 v (1 - v) \qquad (18)$$

The operating principle of the propeller is illustrated in Fig. 16, and that for the thermal jet engine in Fig. 17.

The thrust power equations (17) and (18) demonstrate an important difference between the action of the rocket jet and the other two propulsion systems. Only the rocket can develop useful work at unity velocity ratio. For the others the thrust power is zero at unity velocity ratio because then the thrust is zero.

The exit or leaving loss for the rocket motor P_L (ft-lb per sec) is determined from the absolute exit velocity $c = (w - V)$ of the ejected gases. Thus:

$$P_L = \frac{G}{2g} c^2 = \frac{G}{2g} (w - V)^2 = \frac{G}{2g} w^2 (1 - v)^2 \qquad (19)$$

The ratio of the thrust power P_T to the propulsion power P is a measure of the instantaneous exterior ballistic efficiency of the rocket system, and is termed its propulsion efficiency η_P. Thus:

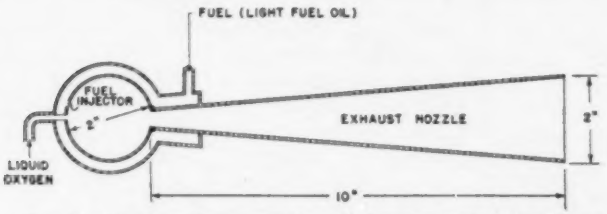
$$\eta_P = \frac{P_T}{P} = \frac{TV}{TV + P_L} = \frac{w^2 v}{\frac{w^2}{2}(1 + v^2)} \qquad (20)$$

Or

$$\eta_P = \frac{2v}{1 + v^2} \qquad (21)$$

Fig. 18 is a plot of the propulsion efficiency of the rocket jet propulsion system as a function of the velocity ratio.

It is seen that the propulsion efficiency of the rocket motor is zero when the system is at rest ($v = 0$). As the flight speed of the system is increased, the velocity ratio



■ Fig. 14 – Regeneratively cooled rocket motor tested by Sänger in 1932 (fuel injected with Bosch diesel-type fuel pump and injection pressure varied from 30 to 150 atmospheres)

increases and so does the propulsion efficiency. When the flight speed equals the effective exhaust velocity ($v = 1$) the propulsion efficiency is unity. At this condition the gases have given up all of their kinetic energy to propulsion work, and the leaving loss is zero.

If the flight speed increases so that the speed of the system is greater than the effective exhaust velocity ($v > 1$), the thrust power is also increased. The increase in output is at the expense of the kinetic energy associated with the moving propellant supply. The propulsion efficiency actually decreases, because all of the energy of the ejected gases is not converted into energy of motion.

To obtain high propulsion efficiency any reaction propulsion system must operate at close to unity velocity ratio. For a rocket motor having an effective exhaust velocity of $w = 6000$ fps the flight speed for unity velocity ratio is approximately 4100 mph. Obviously, the rocket is an inefficient propulsion system except at extremely high flight speeds several times the speed of sound.

Fig. 19 illustrates the effect of flight speed on the instantaneous propulsion efficiency of the rocket motor for different effective exhaust velocities.

The propulsion efficiency equation for either the propeller or thermal jet engine[14] is given by:

$$\eta_P = \frac{2v}{1 + v} \tag{22}$$

Fig. 20 compares the propulsion efficiency of the rocket, engine-driven propeller, and the thermal jet propulsion systems at 20,000-ft altitude.[15]

The thrust and power relationships for the propeller, thermal jet, and rocket jet propulsion are summarized in Table 2. It is seen that only the rocket can develop thrust power at unity velocity ratio.[14]

The function of the rocket motor is to transform the thermochemical energy of the propellants into kinetic energy of the ejected gases.

Let:

E_P = Thermochemical energy per pound of propellants, Btu per lb

h_c = Calorific value of fuel component, Btu per lb

G = Weight flow rate of propellants, lb per sec

G_o = Weight flow rate of oxidizer, lb per sec

G_f = Weight flow rate of fuel, lb per sec

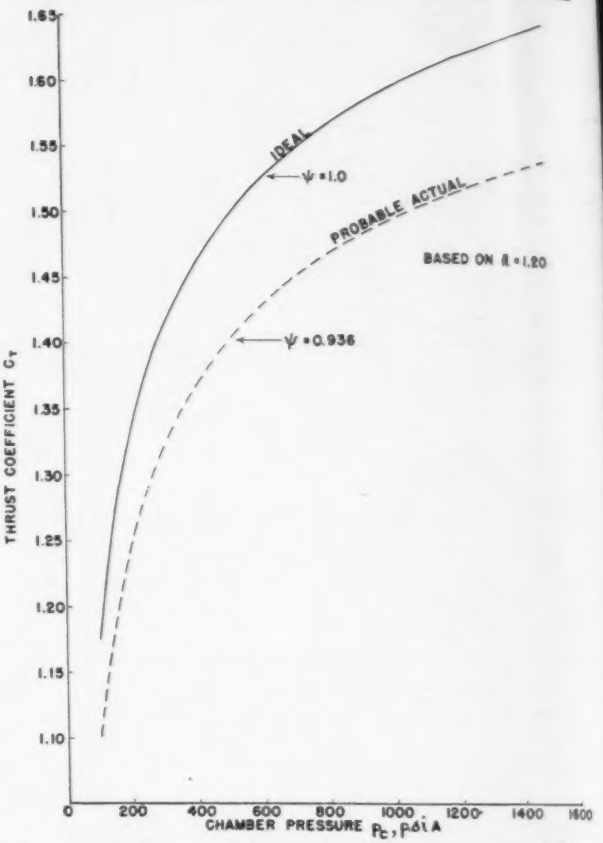
J = 778 ft-lb per Btu

w_u = Ultimate exhaust velocity, fps

w = Effective exhaust velocity, fps

The thermochemical energy (ft-lb per lb) supplied to the

[15] See Journal of Royal Aeronautical Society, Vol. 48, September, October, 1944, pp. 337-389, 397-460: "Aircraft Powerplant – Past and Future," by A. H. Roy Fedden.

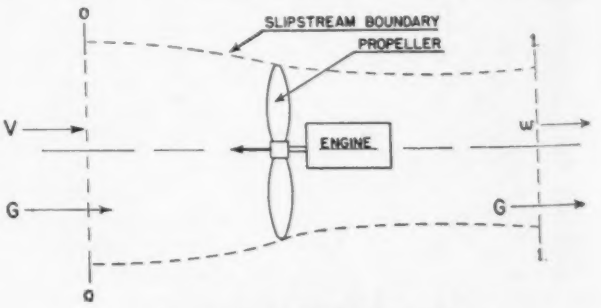


■ Fig. 15 – Effect of combustion-chamber pressure upon thrust coefficient

rocket motor per pound of propellant is given by:

$$JE_P = \frac{JG_f h_c}{G_o + G_f} = \frac{JG_f h_c}{G} \tag{23}$$

If all of this energy could be transformed into kinetic energy, the corresponding velocity of the exhaust gases, called the *ultimate velocity*, would be:



(1) MOMENTUM ENTERING SECTION 0-0

$M_0 = \frac{G}{g} V = mV$

(2) MOMENTUM LEAVING SECTION 1-1

$M_1 = \frac{G}{g} w$

(3) CHANGE IN MOMENTUM PER SEC.

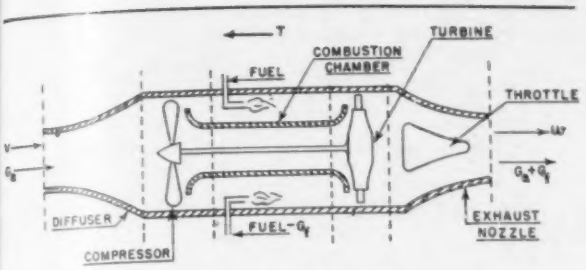
$T = M_1 - M_0 = \frac{G}{g}(w - V)$ LB

AND $\frac{T}{G} = \frac{w}{g}\left(1 - \frac{V}{w}\right) = \frac{w}{g}(1 - v)$

(4) THRUST POWER

$P_t = TV = \frac{G}{g} w^2 (1 - v)v$

■ Fig. 16 – Propulsion action of airplane propeller according to momentum theory



T = THRUST = $\frac{G_a}{g}(w-V) + \frac{G_f}{g}w = \frac{G}{g}(w-V)$

$\frac{T}{G}$ = THRUST/LB. OF AIRFLOW = $\frac{w}{g}(1-\frac{V}{w}) = \frac{w}{g}(1-v)$

P_t = $TV = \frac{G}{g}w(1-v)V$

■ Fig. 17 – Propulsion action of thermal jet propulsion system according to momentum theory

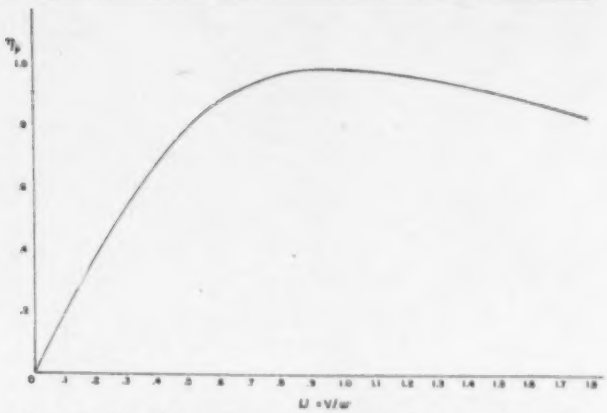
$$w_u = \sqrt{2gJE_P} = 223.7\sqrt{E_P} \qquad (24)$$

The ultimate velocity w_u is a more or less fictitious quantity which cannot be attained even with an ideal rocket motor. The conversion of all of the thermochemical energy of the propellants into kinetic energy if JE_P is measured above absolute zero signifies that there can be no residual heat in the ejected gases; that is, their absolute temperature must be zero. The exit temperature of the gases after adiabatic expansion in the nozzle is determined from:

$$\frac{T_e}{T_c} = \left(\frac{p_e}{p_c}\right)^{\frac{k-1}{k}}$$

Consequently, to obtain $T_e = 0$, the exit pressure p_e and also the back pressure p_a must be zero to avoid overexpansion. This is obtainable only if the rocket motor operates in a vacuum. Furthermore, expansion to zero pressure is possible only if the area ratio F_e/f_t is infinite, which is not possible with a practical motor. The foregoing discussion demonstrates that the ultimate velocity corresponds to the effective exhaust velocity for an ideal rocket motor operating in a vacuum with infinite area ratio.

Due to the aforementioned factors, the various losses, the practical limitations on operating conditions, and dissociation, the actual effective exhaust velocity w is considerably less than the ultimate velocity w_u.



■ Fig. 18 – Propulsion efficiency of rocket jet as function of velocity ratio

The ratio of the actual kinetic energy of the gases $w^2/2g$ to $w_u^2/2g$ is called the internal efficiency η_i. Thus:

$$\eta_i = \frac{w^2}{w_u^2} \qquad (25)$$

Combining equations (24) and (25) gives the following equation for the effective exhaust velocity (fps). Thus:

$$w = 223.7\sqrt{\eta_i E_P} \qquad (26)$$

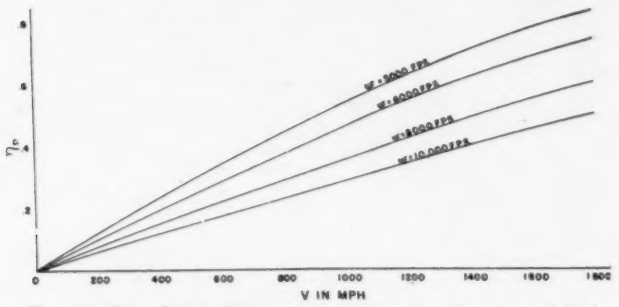
The magnitude of the internal efficiency η_i depends upon the specific propellants, the combustion-chamber pressure, the expansion ratio, the combustion-chamber temperature, and the motor design. In general, the larger the operating pressure ratio the higher the internal efficiency. This means that to obtain the maximum benefits from the thermochemical energy of the propellants the operating pressure for the combustion chamber must be high. This is illustrated in Fig. 21, which presents the maximum expected exhaust velocity for liquid-oxygen gasoline propellant combination under different conditions.[16]

To illustrate the usefulness of the concept of internal efficiency for estimating purposes, consider the reaction between liquid oxygen and gasoline; assuming that gasoline can be represented by octane C_8H_{18}.

Table 2 – Thrust and Power Relationships

Item	Propeller and Thermal Jet	Rocket Jet	Units
Thrust, T	$T = \frac{G}{g}(w - V) = \frac{G}{g}w(1 - v)$	$T = \frac{G}{g}w$	Lb
	$\frac{T}{G} = \frac{w}{g}(1 - v)$	$\frac{T}{G} = \frac{w}{g}$	$\frac{\text{Lb}}{\text{Lb-sec}}$
Exit Loss, P_L	$P_L = \frac{G}{2g}C^2 = \frac{G}{2g}(w - V)^2$	$P_L = \frac{G}{g}C^2$ $P_L = \frac{G}{2g}(w - V)^2$	$\frac{\text{Ft-lb}}{\text{Sec}}$
Thrust Power, P_T	$P_T = TV = \frac{G}{g}w^2(1 - v)v$	$P_T = TV = \frac{G}{g}wV = \frac{G}{g}w^2v$	$\frac{\text{Ft-lb}}{\text{Sec}}$
Propulsion Power, P	$P = P_T + P_L = TV + P_L = \frac{G}{2g}(w^2 - V^2) = \frac{G}{2g}w^2(1 - v^2)$	$P = TV + P_L$	$\frac{\text{Ft-lb}}{\text{Sec}}$
Ideal Propulsion Efficiency, η_P	$\eta_P = \frac{2v}{1 + v}$	$\eta_P = \frac{2v}{1 + v^2}$	———

G = Fluid flow rate, lb per sec; V = True air speed, fps; w = Relative exit velocity of propulsion fluid, fps



■ Fig. 19 – Propulsion efficiency of rocket jet as function of flight speed with different effective exhaust velocities

The stoichiometric equation is:

$$C_8H_{18} + 12.5\ O_2 = 8\ CO_2 + 9\ H_2O \qquad (27)$$

The reaction does not go to completion as indicated above, and the actual reaction can be represented by an equation of the form:

$$C_8H_{18} + 12.5\ O_2 = a\ (CO_2) + b\ (CO) + c\ (H_2) + d\ (OH) + f\ (O) + g\ (H) + \qquad (2^{-})$$

where a, b, c, d, e, f, and g represent the number of mols of each possible constituent in the products formed.

Equation (28), shows that 400 lb of liquid oxygen react with 114 lb of gasoline, or the mixture ratio is $400/114 = 3.5$ The heat of combustion for gasoline above 67 F may be taken to be 19,200 Btu per lb. Hence each pound of propellant mixture entering the rocket motor has $19{,}200/4.5 = 4230$ Btu associated with it. The ultimate exhaust velocity corresponding to this thermochemical energy content is:

$$w_u = 223.7\sqrt{4230} = 14{,}600 \text{ fps}$$

If it is assumed that the maximum attainable internal efficiency in a practical rocket motor is $\eta_i = 0.5$, then the corresponding maximum exhaust velocity to be expected from this reaction is:

$$w = w_u\sqrt{\eta_i} = 14{,}600 \times 0.707 = 10{,}250 \text{ fps}$$

The above value should be regarded as a limiting value which might be attained after considerable research and development. In any case, it will require operation with higher chamber pressures than are currently being used with liquid propellant rocket motors. It is interesting to note, however, that according to W. Ley, Sänger attained effective exhaust velocities ranging from 6600 fps to 11,500 fps with the regeneratively cooled 55-lb thrust motor illustrated in Fig. 14.

Fig. 22 presents the internal efficiency η_i as a function of the effective exhaust velocity for gasoline and ethanol reacting with liquid oxygen in stoichiometric proportions.

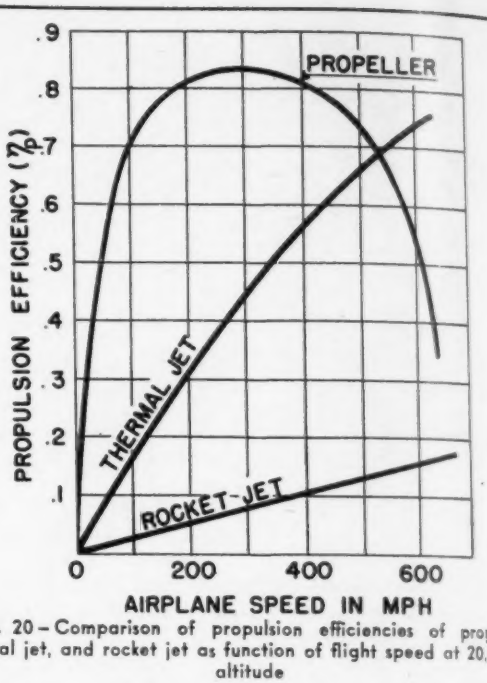
Table 3 presents values of the ultimate exhaust velocity for different liquid fuels reacting with liquid oxygen in stoichiometric proportions. The values of JE_P are measured above 67 F.

The overall efficiency of the rocket jet propulsion system at any instant is given by:

$$\eta = \eta_i \times \eta_P \qquad (29)$$

In this equation the internal efficiency η_i may be considered to be equivalent to the thermal efficiency used as the performance criterion of an internal-combustion engine.

[16] C. F. Brinsmade, unpublished M. S. thesis, M.I.T., 1942.



■ Fig. 20 – Comparison of propulsion efficiencies of propeller, thermal jet, and rocket jet as function of flight speed at 20,000-ft altitude

Under steady state operating conditions its value may be assumed to be constant.

Since the propulsion efficiency η_p varies with the flight speed, the overall efficiency also is a function of the flight speed. To obtain reasonable overall efficiencies the flight speed must be at a high value of the velocity ratio.

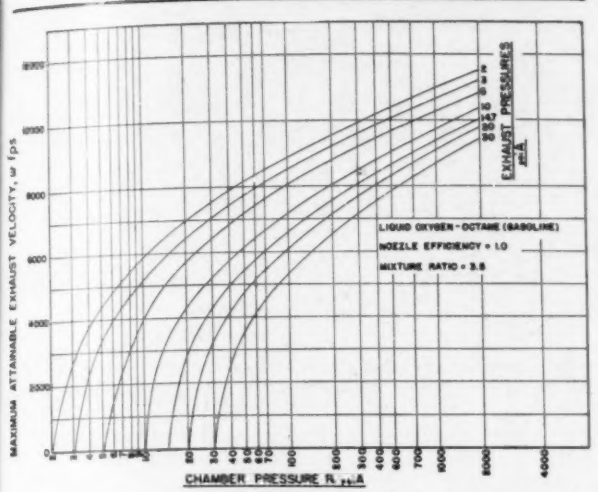
For the engine-driven propeller the internal efficiency, that is the thermal efficiency of the engine, is close to $\eta_i = 0.3$. Its propulsion efficiency at 400 mph, from Fig. 20, is $\eta_P = 0.8$. Hence its overall efficiency in its useful range of operation is approximately 0.24.

A high overall efficiency is not of great practical significance to the rocket designer. His primary concern is to secure the maximum amount of useful work from each pound of propellant contained in the jet propulsion system. This useful energy depends upon the thermochemical energy of the propellants and the efficiency with which it is utilized. The higher the thermochemical energy per pound of propellant, the greater the net propulsion work per pound of propellant, even though the efficiency of utilization is less. As a matter of fact analysis will show that even for low flight speeds, the best propellants from this point of view are those having the highest thermochemical energy content, despite the low propulsion efficiency accompanying their use.

The practical sphere of application of the rocket jet propulsion system can be judged by comparing its fuel consumption rate with that of an engine-propeller system, both operating at the same flight speed V. Assume that both systems are required to propel an airplane against a

Table 3 – Ultimate Exhaust Velocities of Several Fuels Reacting with Liquid Oxygen

Fuel	Chemical Formula	Molecular Weight	Lb O_2 per Lb Fuel	Fuel Heat Value h_c, Btu per lb at 67 F	Propellant Heat Value E_p, Btu per lb	Ultimate Effective Exhaust Velocity w_u, fps
Acetylene	C_2H_2	26.04	3.08	20,770	5100	15,900
Benzine	C_6H_6	78.11	3.08	17,190	4240	14,550
Ethanol	C_2H_5OH	46.07	2.44	11,930	3470	13,150
Gasoline	C_8H_{18}	114.23	3.50	19,245	4280	14,600
Hydrogen	H_2	2.016	8.0	51,608	5750	16,930
Methanol	CH_3OH	32.07	2.0	9,078	3030	12,280


MAXIMUM ATTAINABLE EXHAUST VELOCITY, w fps
CHAMBER PRESSURE P_c, p.s.i.a
EXHAUST PRESSURES p.s.i.a
LIQUID OXYGEN - OCTANE (GASOLINE)
NOZZLE EFFICIENCY = 1.0
MIXTURE RATIO = 3.5

■ Fig. 21 – Maximum attainable effective exhaust velocities for liquid oxygen-gasoline rocket motor at different chamber pressures

constant resistance $D = T$. For the engine, the internal or thermal efficiency may be assumed to be constant and equal to 0.3. The instantaneous overall efficiency as a function of flight speed, based on the propulsion efficiencies of Fig. 20, is presented in Fig. 23.

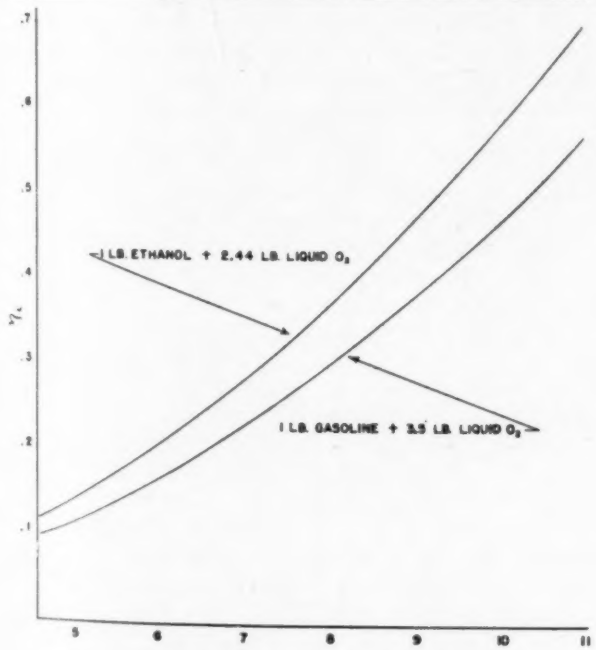
Since the thrust powers are assumed to be the same for both propulsion systems:

$$P_T = TV\,(JE_P)_E\,G_E = \eta_R\,(JE_P)_R\,G_R \qquad (30)$$

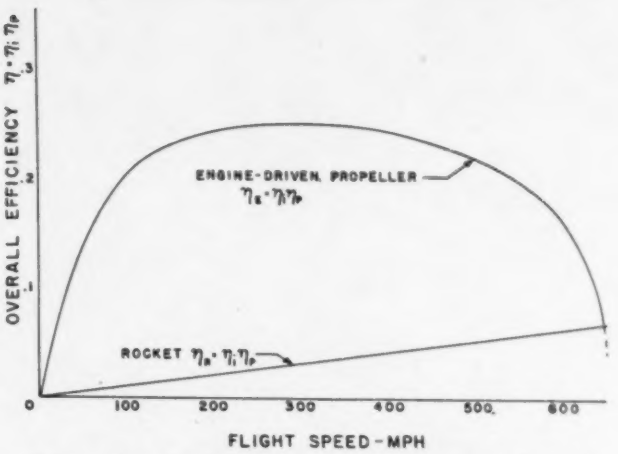
Then the ratio of the rate of propellant flow G_R to the engine fuel consumption G_E is given by:

$$\frac{G_R}{G_E} = 5.76\,\frac{\eta_E}{\eta_R} \qquad (31)$$

If it is assumed that the effective exhaust velocity for the



■ Fig. 22 – Internal efficiency plotted against effective exhaust velocity for ethanol and gasoline used with liquid oxygen in stoichiometric proportions


OVERALL EFFICIENCY $\eta_t \cdot \eta_i \eta_p$
ENGINE-DRIVEN PROPELLER
$\eta_E \cdot \eta_i \eta_p$
ROCKET $\eta_R \cdot \eta_i \eta_p$
FLIGHT SPEED - MPH

■ Fig. 23 – Overall efficiency of engine-driven propeller and rocket jet as function of flight speed

η_E = Overall efficiency for engine-driven propeller
η_R = Overall efficiency for rocket jet propulsion

rocket motor is 6400 fps, the value reported for the German V-2 missile, the internal efficiency for the rocket motor is 0.237. Hence, its instantaneous overall efficiency is:

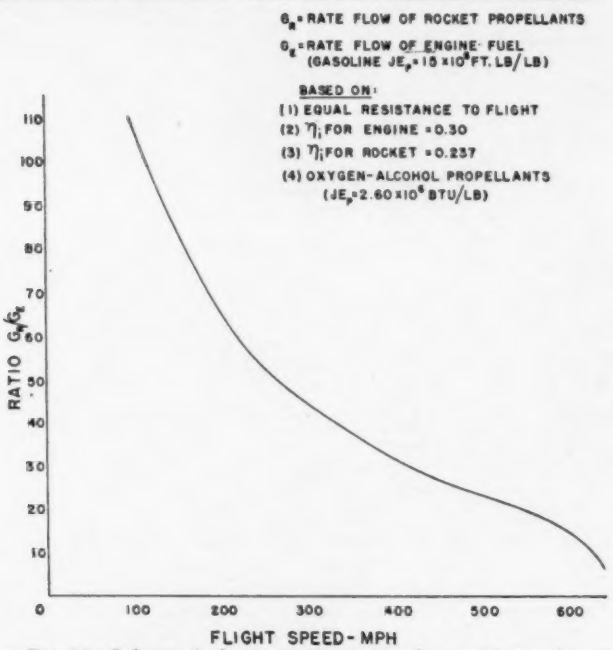
$$\eta_R = 0.237\,(\eta_P)_R = 0.237\,\frac{2v}{1 + v^2}$$

The results of the calculations of the relative fuel consumption rates are presented in Fig. 24. It is apparent from the figure that at all flight speeds up to the limit of operation for the propeller, the rocket consumes propellants at several times the rate that the engine consumes fuel. On a purely propellant consumption basis, the rocket cannot compete with the conventional system. Its application to aircraft must, therefore, be based upon the major demands being high rate of climb, high-altitude operation, and higher than normal flight speeds. These advantages can be obtained with the rocket, if the extremely short endurance is tolerable. The application of the rocket motor as the sole propulsion system for aircraft is, consequently, limited to high-speed intercepter-type military craft. The ME-169 used by the Germans during the final weeks of the European War and referred to earlier was a rocket propelled airplane. It is reported to have attained higher speeds than any conventionally propelled airplane used in the European battle front, but its duration was about 10 min.

In this country the main application of rockets to aircraft, apart from their use as weapons, has been as an auxiliary to the propeller to reduce take-off distance and/or time. The majority of applications during the war was to flying boats where the added thrust of the jato (jet assisted take-off) units made it possible to take off under conditions such that without them successful take-off would have been problematical. This increased the usefulness of these aircraft for rescue work in forward combat areas. Jato units have also been applied to both carrier- and land-based aircraft. As is to be expected, reductions in both take-off distance and time resulted from their use. For an analysis of the take-off considerations for seaplanes, see Stout.[17]

Fig. 25 illustrates pictorially the improvement in the sea-level take-off characteristics of a DC-3 transport effected

[17] See *Aviation*, Vol. 44, October, November, December, 1945, pp. 137-141, 140-146, 163-166: "Take-Off Analysis for Flying Boats and Seaplanes," by E. G. Stout.



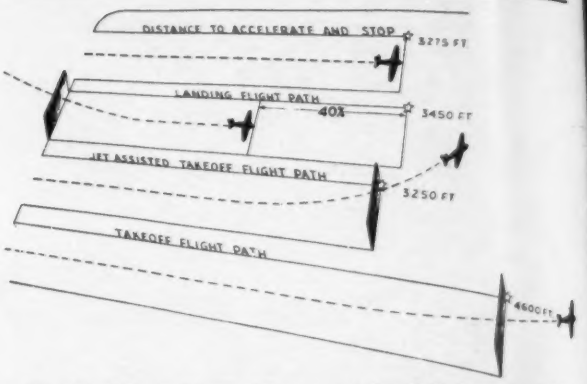
■ **Fig. 24 – Relative fuel consumptions of rocket and internal-combustion engine as function of flight speed**

by equipping it with a 1000-lb jato unit having a duration of 14 sec. Fig. 26 is a similar comparison applying to take-off at an elevation of 6000 ft. It is seen that the distance to clear a 50-ft obstacle is reduced from 4600 ft to 3250 ft at sea level, and from 5800 ft to 3850 ft at 6000-ft elevation.

Two types of rocket jet propulsion units have been employed for assisted take-off purposes. Those employing restricted burning solid propellants, and those utilizing liquid propellants. In their wartime use, the solid propellant units found the greatest favor, due to ease of installation and fewer handling and logistic problems.

Fig. 27 illustrates a 1500-lb thrust, 40-sec duration liquid propellant jato system. The unit is controlled from a single electric switch operated by the pilot. The switch actuates an electric control valve, thereby causing an inert pressurizing gas to communicate with the propellant tanks. The pressure of the inert gas is controlled by a regulating valve to maintain the same constant gas pressure on the liquid in each propellant tank. The pressurizing gas is also utilized for actuating the propellant flow-control valves located upstream to the injection device for feeding the propellants into the rocket motor. No ignition system is employed, since the propellants used with this unit react spontaneously upon contact with each other. The characteristic thrust-time curve obtainable from this unit is illustrated in the left-hand upper corner of the figure. The thrust-time curve is obtained from the standard static firing tests on unit before accepting it for delivery to the user.

Liquid propellant jato units have been constructed so that they can be mounted either as fixed installations in the aircraft, or so that they can be dropped by parachute after use. Reliable liquid propellant rocket motors have been developed that have operated without difficulty for continuous periods, limited only by the capacities of the propellant tanks. These motors are light in weight and the combustion chamber and nozzle are cooled regeneratively by circulating one of the propellants around them prior to injecting the fluid into the motor. A rocket motor of this type developing 1500- to 2000-lb thrust can be built



■ **Fig. 25 – Effect of jet assistance on take-off flight path of DC-3, SIC3G, if certificated under transport category – sea-level conditions (calculated)**

1. Take-off and landing flight paths as defined by CAR 04.750T
2. Standard atmosphere, no wind, hard surface runway
3. Gross weight at take-off is 26,200 lb or maximum permitted by take-off climb requirement
4. Gross weight at landing is 25,200 lb or maximum permitted by structural requirement
5. Jet assistance – 1000 lb of thrust for 14 sec

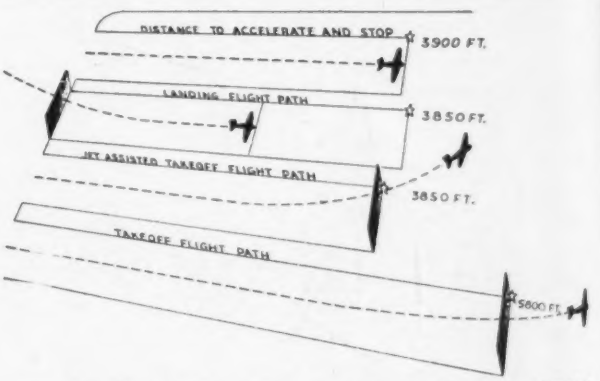
at a weight slightly less than 45 lb. For security reasons, the design details of this type of motor cannot be released nor can information be presented regarding the developments concerned with propellants or pressurizing means.

Fig. 28 is a photograph of a PBM seaplane taking off with two droppable liquid propellant units.

Fig. 29 presents a cross-section through a typical solid propellant jato unit that delivers 1000 lb for 12 sec. Its operation is initiated by the igniter which is fired electrically.

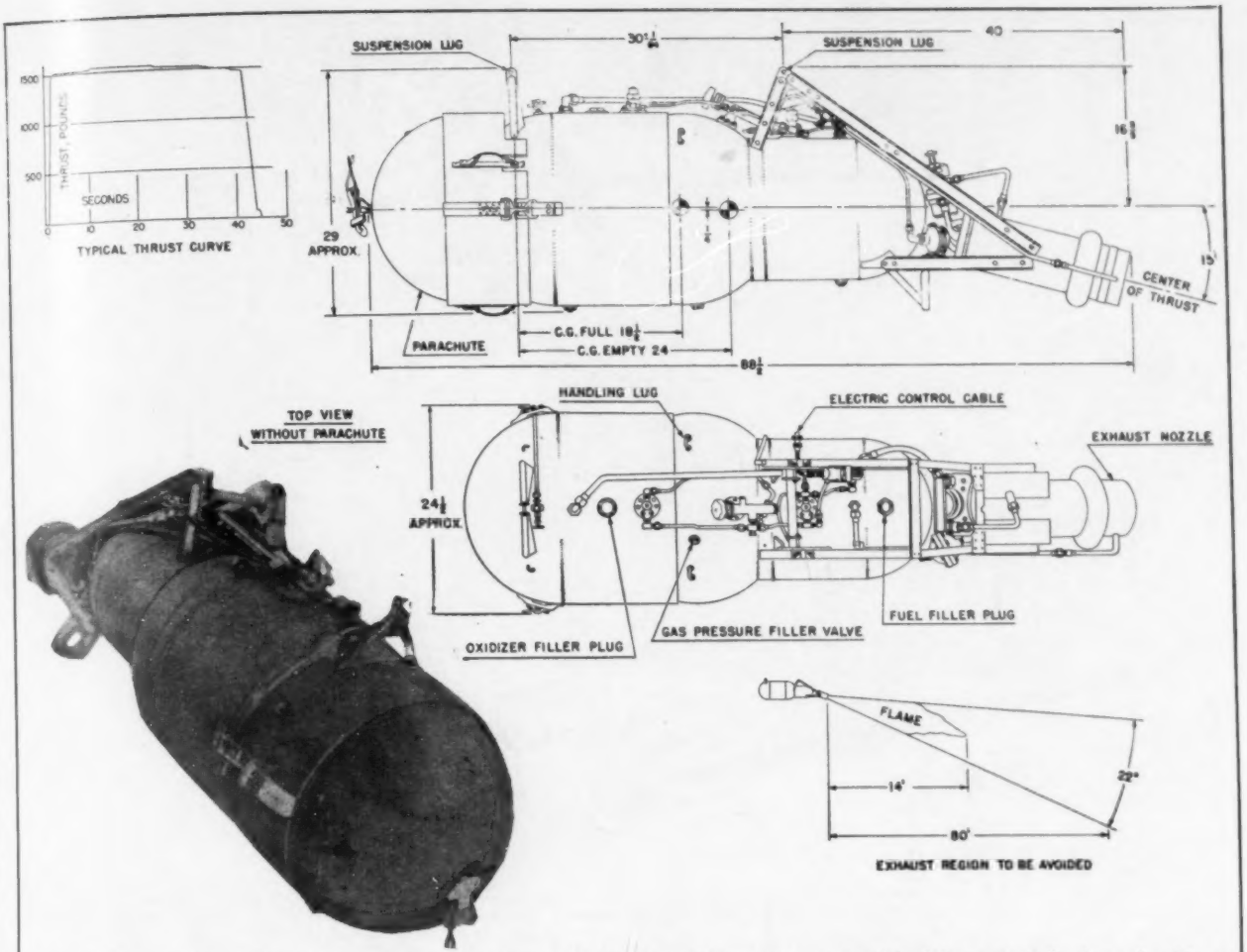
Fig. 30 shows a Lockheed P-80 airplane taking off with these units arranged to supply additional thrust.

The rocket motor has been developed to the stage where it can be applied as the powerplant for short-duration, high-speed military aircraft. Its usefulness for assisted take-off and as the propulsion system for long-range missiles, such as the German V-2, has been established, and no doubt it will find application in the future development of pilotless aircraft for high-altitude, high-speed flight. Other



■ **Fig. 26 – Effect of jet assistance on take-off flight path of DC-3, SIC3G, if certificated under transport category – 6000-ft altitude (calculated)**

1. Take-off and landing flight paths as defined by CAR 04.750T
2. Standard atmosphere, no wind, hard surface runway
3. Gross weight at take-off is 24,200 lb or maximum permitted by take-off climb requirement
4. Gross weight at landing is 24,400 lb or maximum permitted by structural requirement
5. Jet assistance – 1000 lb of thrust for 14 sec



■ Fig. 27 – Cross-section and exterior appearance of droppable liquid propellant jato unit which delivers 1500-lb thrust for 38 sec

applications where its advantages may be of value are for torpedo drive, hydrobombs, and sounding rockets.

The high-temperature, high-pressure gas generating capabilities should also find fields of application.

For example, the gases generated by reacting rocket propellants can be utilized as the working fluid for operating auxiliary turbines for the short periods required for starting rotating machinery. It can be said that for any service where a temporary source of high-pressure gases can be employed for useful purposes, the possibilities of a rocket system are worthy of investigation.

In conclusion it can be said that as far as airplane propulsion is concerned, jet propulsion devices, such as the thermal jet engine and the rocket motor, have reached the development stage where they must be given serious consideration as propulsion systems, especially where high-speed or high-altitude flight is of paramount importance. The thermal jet engine has now demonstrated its ability to make high flight speeds a possibility. For shorter duration higher speed flight at higher altitudes, the rocket jet has demonstrated its potentialities.

The author expresses his thanks to T. B. Swanson, Aerojet Engineering Corp., for his assistance in making several of the calculations.

Appendix

Symbols

T = Thrust, lb

T_c = Temperature of gases in combustion chamber, R

p_e = Absolute pressure of gases at nozzle exit section, psi (absolute)

p_a = Atmospheric back pressure, psi (absolute)

p_c = Combustion-chamber pressure, psi (absolute)

f_t = Throat area, sq in.

F_e = Exit area of nozzle, sq. in.

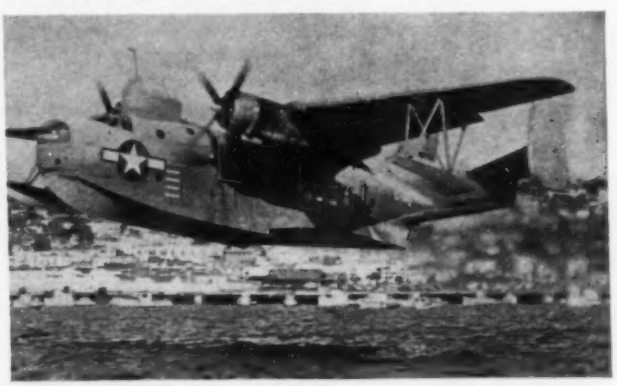
$G = G_o + G_f$ = Rate of propellant consumption, lb per sec

G_o = Rate of oxidizer consumption for rocket, lb per sec

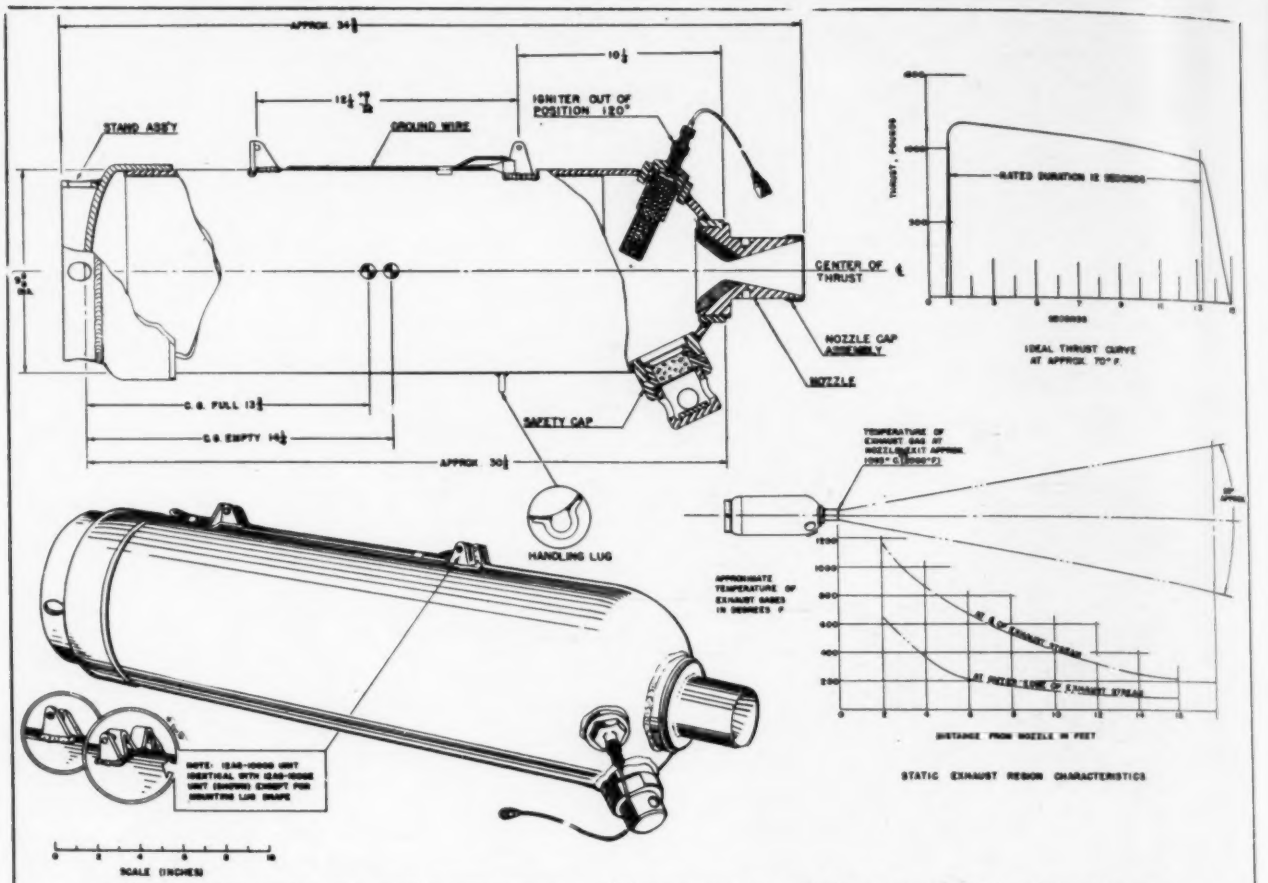
G_f = Rate of fuel consumption for rocket, lb per sec

G_E = Rate of fuel consumption for engine-driven propeller, lb per sec

G_R = Propellant flow to rocket motor, lb per sec



■ Fig. 28 – PBM seaplane taking off with liquid propellant jet propulsion units



■ Fig. 29 – Cross-section and exterior appearance of solid propellant jato unit that delivers 1000 lb of thrust for 12 sec

$a = \sqrt{gkRT}$ = Acoustic velocity

w = Effective exhaust velocity for rocket, fps

w_u = Ultimate exhaust velocity for rocket propellants, fps

V = Flight speed, fps

$c = w - V$ = Absolute exit velocity for gases, fps

c_e = Velocity of gases at exit section relative to the nozzle walls, fps

g = Acceleration due to gravity – 32.174 ft per sec^2

$I = w/g$ = Specific impulse, lb per lb-sec

$J = 1/A$ = 778 ft-lb per Btu

h_c = Calorific value of fuel, Btu per lb

$k = c_p/c_v$ = Specific heat ratio

c_p = Specific heat at constant pressure, Btu per lb-F

c_v = Specific heat at constant volume, Btu per lb-F

$R = 1545/m$ = Gas constant [where p is in psf (absolute)

m = Molecular weight

P = Propulsion power, ft-lb per sec

■ Fig. 30 – Lockheed P-80 airplane taking off with solid propellant jato units

$P_T = TV$ = Thrust power, ft-lb per sec

$P_L = \frac{G}{2g} c^2$ = Exit loss, ft-lb per sec

E_p = Thermochemical energy of rocket propellants, Btu per lb

$(JE_P)_E$ = Thermochemical energy of engine fuel (gasoline) = 15×10^6, ft-lb per lb

$(JE_P)_R$ = Thermochemical energy of rocket propellants (oxygen and ethanol in the stoichiometric ratio) = 2.6×10^6, ft-lb per lb

D = Drag, lb

ϕ = Velocity coefficient = $\frac{\text{Actual velocity}}{\text{Ideal velocity}}$

$\psi = \frac{\text{Actual thrust coefficient}}{\text{Ideal thrust coefficient}}$

$v = w/V$ = Velocity ratio

λ = Correction factor for nozzle divergence angle

α = Nozzle divergence angle

η_n = Nozzle efficiency

η_i = Internal efficiency

η_P = Propulsion efficiency

η = Overall efficiency

η_R = Overall efficiency for rocket

η_E = Overall efficiency for engine-driven propeller

Subscripts

e = Exit section

E = Engine-driven propeller

f = Fuel

o = Oxidizer

R = Rocket

DAMPING IN SUSPENSIONS

by
B. E. O'CONNOR
Houde Engineering Division
Houdaille-Hershey Corp.

A RATIONAL discussion of the effect of damping in suspensions requires a criterion of the qualities of a good ride. Unfortunately, the exact nature of this criterion is debatable. Cain[1] presents a comfort index developed at Purdue University which is a function of the acceleration and the frequency. Den Hartog[2] states that the change in acceleration is a criterion of riding comfort. The point need not be argued here. Instead, the effect of damping will be discussed with regard to velocity, acceleration, and change in acceleration of the vehicle body.

There are several types of damping which affect the motions of a vehicle body in different manners. Six of the more important types are:

1. Coulomb damping, which is the mathematician's idealization of dry friction damping. It is defined as a constant resistance to motion, regardless of relative velocity. It is rarely found in practice as the result of rubbing between two dry surfaces. It is, however, produced in fluid dampers in which the pressure is held constant by means of a relief valve.

2. Actual dry friction, which is characterized by a resistance to motion that is inversely proportional to the velocity. It occurs where there is relative motion between two dry surfaces in contact. The force resisting motion is a function of the force holding the two surfaces together, the materials in contact, and the nature of the surfaces, as well as the velocity.

3. Viscous damping, which is characterized by a resisting force that is linearly proportional to the velocity. Its name is derived from the definition of absolute viscosity. It is the only type of damping which can be dealt with in a straightforward manner in problems concerned with harmonic motion. In practice, it is found where there is relative motion between two well lubricated surfaces, and where a viscous fluid is forced through a relatively long passage of small cross-sectional area.

4. Degenerate viscous damping, which is characterized by a resistance to motion that is proportional to a power less than one of the velocity. It is frequently found in practice, but rarely if ever in the literature. Its characteristics are derived from the deviation of viscosity in most actual fluids from the definition of absolute viscosity.

5. Hydraulic damping, which is characterized by a resisting force that is proportional to the second power of the velocity. This relationship is attained by forcing fluid of low viscosity through a sharp edged orifice.

6. Hysteresis damping, which is a function of the stress in a material, the energy dissipated per cycle being proportional to the area of the loop formed by the deviation from Hooke's law on a stress-strain diagram of a complete stress reversal. In practice, it is assumed to be proportional to a power greater than one of the stress for a given object.

All six types of damping are present to some degree in the majority of suspensions.

Most shock absorbers incorporate blow-off valves which contribute degenerate viscous damping as they begin to open and coulomb damping at still higher velocities.

Dry friction is present in poorly lubricated leaf springs, shackles, and many other points of nonlubricated rubbing surfaces, such as doors.

Viscous damping or degenerate viscous damping is present in well lubricated leaf springs and shackles and in shock absorbers using a viscous fluid and relatively small orifice.

Hydraulic damping is found in shock absorbers in which the size of the orifice is large in proportion to the viscosity of the fluid.

Hysteresis damping is present in all parts of the vehicle which undergo deformation and is of special importance in rubber tires.

THE various types of damping present in vehicle suspensions and their effect on the ride are discussed by Mr. O'Connor, who compares the characteristics of the various types of damping with the mathematically adaptable viscous damping.

The ideal damper and the present-day practical approach are considered.

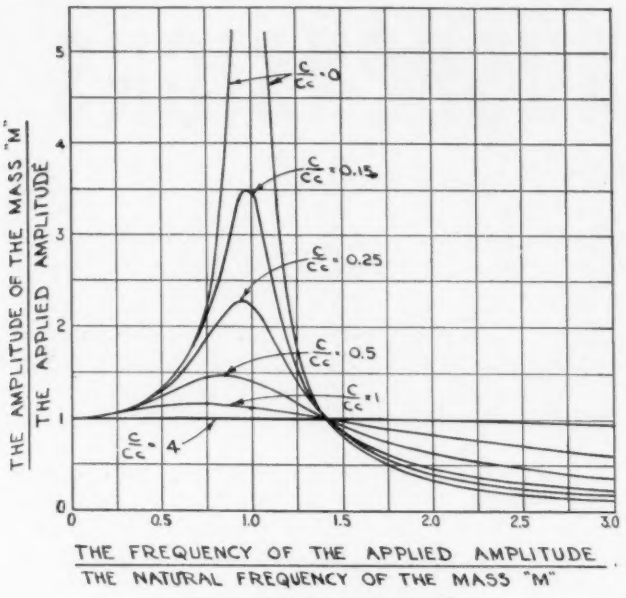
The discussion is maintained on a nonmathematical plane throughout, as the author believes that the mathematics of vibration theory is ably covered in numerous texts on the subject.

THE AUTHOR: B. E. O'CONNOR (M '45) joined the Houde Engineering Division of the Houdaille-Hershey Corp. in 1935 as a draftsman, and has since then been appointed chief draftsman in 1942 and assistant chief engineer in 1944. A graduate of Newark Technical School, Newark, N. J., Mr. O'Connor's experience includes work with shock absorbers and general vibration control, covering shimmy of aircraft nose wheels and torsional vibration in the crankshafts of internal combustion engines.

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 9, 1946.]

[1] See "Vibration of Rail and Road Vehicles," by B. S. Cain. Published by Pitman Publishing Corp., New York, 1940.

[2] See "Mechanical Vibrations," by J. P. Den Hartog. Published by McGraw-Hill Book Co., New York, 1940.



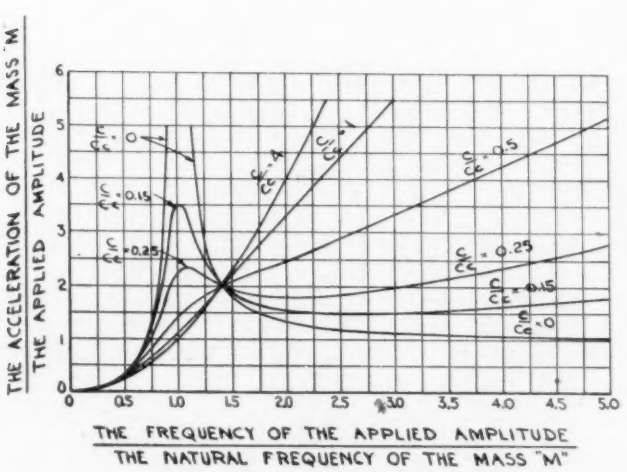
■ Fig. 1 – Amplitude of mass M

Although there is inherent damping in all suspensions, usually it is not of sufficient magnitude to provide proper control. Also, it is not necessarily the type of damping which contributes to the optimum ride.

Rational design then dictates that the inherent damping in a suspension be held to a minimum and separate dampers with controllable characteristics be used.

For purposes of simplicity, the discussion will be confined to simple one- and two-degree of freedom systems. The principles involved, however, apply equally well to the more complex system of a vehicle suspension.

The phenomenon of resonance dictates a considerable amount of damping in the system, as illustrated by Figs. 1 and 2. These figures are developed for the simple system shown in Fig. 3, where a mass is connected to a harmonically alternating vertical displacement through a spring and a viscous damper in parallel. This is analogous to the system of a vehicle body supported on an axle by means of a spring with a shock absorber connected between the body and the axle, the axle being subjected to a harmonically varying vertical displacement.



■ Fig. 2 – Acceleration of mass M

Fig. 1 plots $\frac{\text{amplitude of the mass } M}{\text{applied amplitude}}$ as a function of $\frac{\text{frequency of the applied amplitude}}{\text{natural frequency of the mass } M}$ for various amounts of damping. In order to render the curve dimensionless the damping is expressed as C/C_c, where C is equal to lb force per unit of velocity of the damper, and C_c is the critical damping, which is the minimum value of C at which the displacement of the mass M from the equilibrium position will not result in an oscillation but merely in the return of the mass to the equilibrium position.

Fig. 2 plots $\frac{\text{acceleration of the mass } M}{\text{applied amplitude}}$ as a function of $\frac{\text{frequency of the applied amplitude}}{\text{natural frequency of the mass } M}$.

Since Fig. 1 deals with the amplitude of the mass M, and Fig. 2 deals with the acceleration or second derivative of the amplitude with respect to time, the shape of the curves for the succeeding derivatives may be visualized as having the same general characteristics as Figs. 1 and 2.

Figs. 1 and 2 show that, for values of the frequency ratio of less than $\sqrt{2}$ a large amount of damping is beneficial. It is seen that for the case of zero damping the amplitudes and accelerations approach infinity. It is also seen that for values of the frequency ratio greater than $\sqrt{2}$ damping contributes to greater values of amplitude and acceleration.

The mathematical theory on which Figs. 1 and 2 are based may be found in any standard text on mechanical vibration. The theory is based on an applied harmonic displacement. Obviously, the probability of having a pure harmonic displacement applied to the axle of a vehicle is remote. However, by means of the Fourier analysis, any repeated pattern of nonuniformities may be resolved into a series of sine waves. The first sine wave has a frequency determined by the time necessary to travel from one irregularity or group of irregularities to the next; the second sine wave has a frequency of twice the first; the third sine wave has a frequency of three times the first, and so on, to infinity. It is seen that, if the speed of a vehicle is such as to make the frequency of any of these sine waves the same as the natural frequency of the vehicle body, resonance will result.

Fig. 4 shows graphically the results of the Fourier analysis of the path of an axle as a wheel of constant rolling radius passes over a series of evenly spaced elliptical bumps.

Damping is also desirable as a means of rapidly destroying the free vibrations set up by an occasional bump on an otherwise smooth contour.

The quantity of damping required in a suspension for optimum ride control is directly proportional to the mass and the natural frequency. It is seen then that a vehicle with relatively soft springs requires less damping than a vehicle with relatively stiff springs.

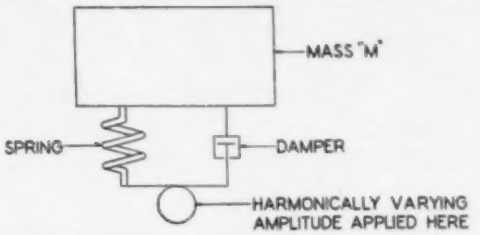
Although the basic theories both for resonant conditions and free vibrations are developed for the case of viscous damping, the curves of Figs. 1 and 2 are reasonably accurate for damping other than viscous for the amounts of damping found in ordinary suspensions. It is only neces-

sary that the damping dissipate the same energy over a complete cycle as would a viscous damper. Unfortunately, a damper with characteristics other than viscous can be made to dissipate the same amount of energy as a viscous damper under one set of conditions only. A so-called equivalent viscous damper set up for one frequency and amplitude will have the value of a different amount of viscous damping for any other frequency or amplitude. Thus, an equivalent viscous coulomb damper at a given forced amplitude and frequency of the axle will have the effect of a larger viscous damper for smaller amplitudes or lower frequencies, and the effect of a smaller viscous damper for larger amplitudes or higher frequencies. This relationship is also true for the cases of dry friction and degenerate viscous damping. The inverse is true of hydraulic damping. The equivalent viscous damping capacity of a hysteresis damper will be small for amplitudes less than the design amplitude, small for frequencies greater than the design frequency, large for amplitudes greater than the design amplitude, and large for frequencies less than the design frequency.

In general, the concept of equivalent viscous damping is applicable to the comparison of the various types of damping under transient conditions if the term velocity is used to replace the term frequency. Thus, for equivalent dampers, the maximum forces imparted to the body by a coulomb or degenerate viscous damper will be less at high velocities and greater at low velocities than the forces imparted by a viscous damper, and again the inverse is true of the hydraulic damper.

An important consideration is the phase relationship between the damper force and the resultant of the spring force and the weight of the body. For the case of coulomb damping, the maximum values of these forces occur simultaneously and must be added directly. The maximum viscous or hydraulic damper forces occur at the maximum velocity, which cannot be concurrent with the maximum resultant of the spring force and body weight.

The damping out of free vibrations presents a different picture. In this case, the axle is relatively steady, while the body moves up and down. It can be shown analytically



■ Fig. 3 – Simple system having mass connected to harmonically alternating vertical displacement through spring and viscous damper in parallel

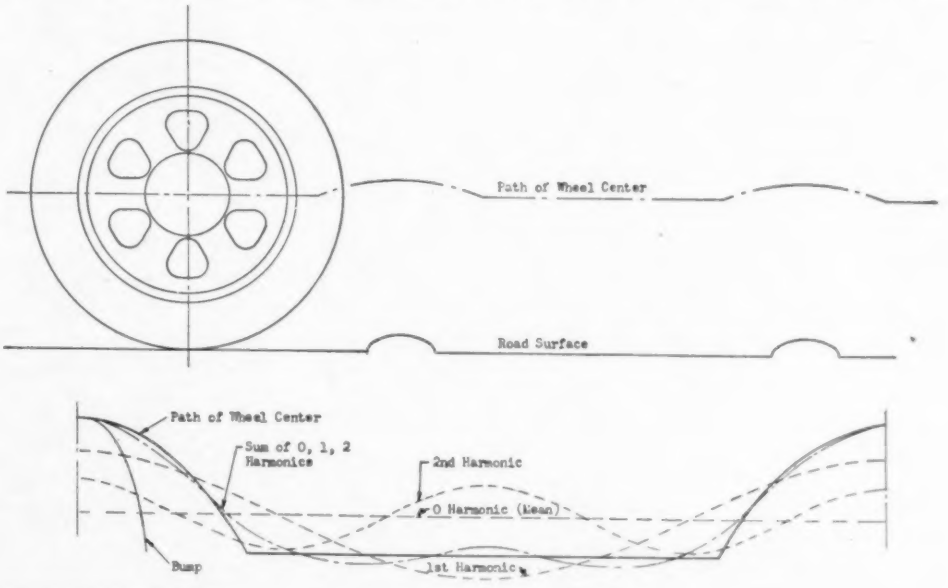
that the effect of damping is to reduce the amplitude of each succeeding oscillation, and to reduce the accelerations and changes in acceleration for all motions except the initial displacement.

In the light of the foregoing discussion, the following conclusions can be drawn:

1. The factors influencing the quality of a ride which are affected by the damping in the system are the acceleration and the change in acceleration of the vehicle body.
2. Damping is essential in the vicinity of resonance and for destroying free vibrations, and objectionable for all other conditions.
3. Viscous damping tends to amplify road irregularities in proportion to their severity.
4. Coulomb and degenerate viscous damping tend to amplify the effect of mild irregularities.
5. Hydraulic damping tends to amplify mild irregularities to a lesser extent than does viscous damping and to amplify severe irregularities to a greater extent.

Assuming that the hysteresis damping in the tires is sufficient to prevent large resonant amplitudes of the unsprung mass of a vehicle, the ideal damper is one which damps only free body vibrations and resonant phenomena.

An approach to this ideal is the inertia-control shock absorber. Since the two conditions under which damping is beneficial involve body motion and all other conditions involve only axle motion initially, it is seen that a damper which is controlled by motion of the body will approach



■ Fig. 4 – Fourier analysis of axle motion as wheel passes over series of evenly spaced elliptical pumps. Vertical scale of lower figure four times horizontal scale

■ Fig. 5 – Direct-acting shock absorber

the ideal if motion of the body introduces damping to the system at the end of the first cycle of body motion.

The inertia-control shock absorbers with which the writer is familiar do not take full advantage of the inherent possibilities, as this operation is ideal only for the case of a rise not followed by a drop and the case of a drop which may be followed by a rise but not by another drop. This pattern is accomplished by having the motion of the shock absorber always free as the axle approaches the body. The valve is set by inertia for damping motion of the body away from the axle when the body is accelerated. The ideal inertia-controlled damper will give optimum results for a rise either followed or not followed by a drop, or a drop either followed or not followed by a rise, but neither type of inertia control damper will act ideally when a bump is closely followed by another bump, unless the spacing is such as to cause resonance.

Another approach to the ideal damper is a frequency sensitive damper. The basis for this approach is the fact that both free and resonant vibrations occur at the same frequency, so that a damper which damps only in the vicinity of the natural frequency of the body approaches the ideal.

At the present time, ideal dampers are too complex for general use and a practical approach is made to the problem.

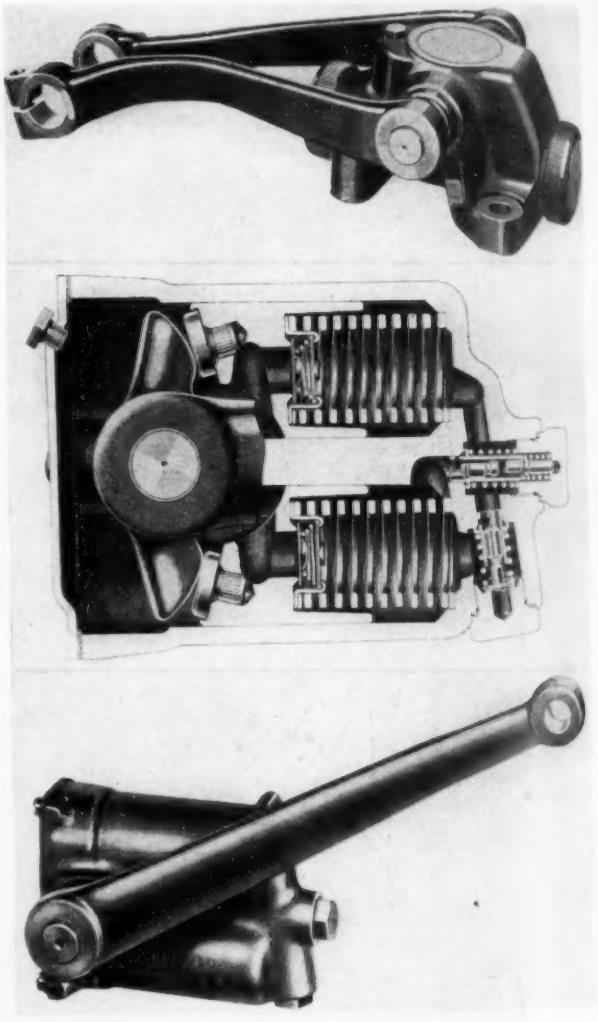
There are three types of shock absorbers in general use, namely, the rotary vane, the direct-acting, and the cam-actuated piston. Actually, they differ mainly in structural detail. The principles involved are the same. Examples of the three types are shown in Figs. 5, 6, and 7.

Basically, the resistance to motion is created by forcing a fluid through an orifice. Relief valves are provided to limit the maximum resistance in both directions. In most cases, the resistance as the axle and body come together is much less than the resistance when the axle and body part.

The reason given for this practice is that with a damper in the system there is no limit to the body accelerations which might be encountered in rising over an obstacle, whereas the accelerations imported to the body by a dropping axle cannot exceed the acceleration of gravity.

The amount of damping in shock absorbers is usually of the order of 0.15 to 0.2 of critical damping.

The energy dissipated by a damper appears as heat. The amount of energy dissipated is a function of the damping constant, the frequency, and the square of the amplitude. The average automobile shock absorber oscillating at a frequency of 85 vibrations per min through an amplitude of 2 in. above and below the equilibrium position will dissipate approximately 7 Btu per min. This is equivalent to 0.16 hp per shock absorber. This power must be supplied by the engine. When traveling over extremely rough roads, the temperature rise in a rotary vane or cam-actuated shock absorber is approximately 35 F, and the rise in the direct-acting type is approximately 90 F. The reason for



■ Fig. 6 – Cam-actuated piston type of shock absorber

this difference is the method of attachment to the vehicle. The direct-acting type is held at either end by a rubber bushing which acts as an insulator so that the majority of the heat must flow from the damper body to the surrounding air by means of convection and radiation. The bodies of the other two types are held in intimate contact with the frame of the vehicle and heat flows to the chassis by conduction.

The effect of temperature on the characteristics of a given damper depends on the fluid used in the damper and the type of valving. Until recently, it was necessary to use a much more viscous fluid in the rotary vane type of damper than was necessary in the other two types, but this has been overcome by the development of internal seals. In general, changes in temperature produce greater changes in the characteristics of dampers using high viscosity fluids than is the case for dampers using low viscosity fluids.

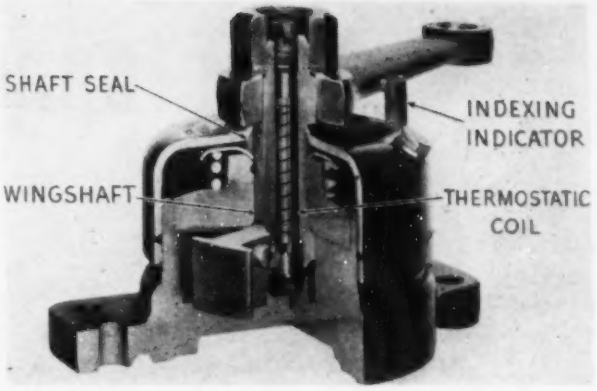
The new silicone fluids show remarkably small change in viscosity for changes in temperature. For this reason, they are very excellent shock absorber fluid but at present the cost prohibits their use for this purpose.

Silicone fluids have been used successfully by the writer to produce damping by merely shearing a film of fluid between two surfaces. Viscous or degenerate damping of sufficient magnitude for automotive shocks has been produced. Torsional crankshaft dampers using this principle have been successfully tested on two large diesel engines and one automobile engine.

The maximum pressures developed in conventional shock absorbers are of the order of 400 psi in the direct-acting type and 1500 psi in the cam-actuated piston and rotary vane type.

The approach to the damping problem has been more experimental than theoretical in the past. The simple principles stated herein have been used to indicate the direction of the experimental approach.

It is hoped that in the future as the inherent damping in vehicles is decreased, theory may be used to indicate quantitively the proper amount and type of damping which will provide a ride which may be called positively "the best possible ride with a given suspension."



■ Fig. 7 – Rotary valve type of shock absorber

DISCUSSION

Asserts Damping Specs Should Be Predicated on Average Conditions

– R. E. LEWTON
Chrysler Corp.

USING the mathematical approach the author derives the nature of the damping requirements for resonant and free vibrations of the car body, and if any confirmation were needed we could draw from experience to say that his conclusions appear to be in substantial agreement with experimental observation. He has, however, treated the case in which "the hysteresis damping in the tires is sufficient to prevent large resonant amplitudes of the unsprung mass of the vehicle." The more usual case in which there is *insufficient* damping in the tires is also one in which we are vitally interested.

In the rear of the present-day passenger car a weight of 300 to 400 lb is suspended between two flexible supports: the tires and the chassis springs. Relatively slight harmonic movements of the unsprung mass excited by irregularities of the road surface tend to be continued or even amplified by energy coming from the powerplant. Modifications or complete interruptions of the road and tire contact once each cycle result in a feed of energy from the powerplant which is in phase with the energy initiating the wheel movement. The resulting "axle dance" or "axle chatter" is very annoying to the driver both because of the obvious commotion under the car and the lack of continuous, stable contact with the road surface. It contributes to traction difficulties in soft sand or on corduroy roads where it may be necessary to throttle down until the vibration stops. On a rough road, particularly on a curve, the car may take a dangerous side hop. Also the condition of nonuniform tire contact results in successive accelerations and decelerations of the tires and, therefore, increased tread wear.

It would be of considerable value if an analysis could be made along the lines suggested in the paper showing where and how to apply the proper damping for this mass. Much progress has been made experimentally, but there is still much to be accomplished. Currently, a large share of damping which should (if it could) be located between the unsprung mass and the road is placed in the springs, shock absorbers, and other connections between the unsprung weight and the car body – in other words: in the same damper as that used to damp the free and resonant vibrations of the car body. Since the requirements for both systems are never identical the choice is always a compromise not properly adapted to either purpose. Furthermore, it will be noted that reactions from the damper are taken by the car body, which is to say that a fraction of every impulse which gets through the tire is transmitted directly to the body without benefit of an intervening spring. In all cases this sets up vibrations in the body which often are of objectionable intensity and if the damping is too severe a harsh ride results.

The situation at the front of the vehicle is not as aggravated as at the rear, since there is no power application to deal with, and the unsprung weight is considerably less. Due to mechanical difficulties in the arrangement of the steering gear connections, however, damping of the unsprung mass is a factor in maintaining a steady steering wheel and is given careful attention during selection of the shock absorber valving.

Another use of damping is in the steering linkage itself to prevent shock loads from being transmitted to the steering wheel. Both hydraulic and friction dampers have been considered. Plain spindle bearings are sometimes selected over other types because of the friction they provide.

Hydraulic dampers are effectively used on some models to resist body sway and to damp lateral movements of the rear axle. The interleaf friction of chassis springs on many models also serves to damp lateral movements.

Mr. O'Connor emphasizes one of the less understood points – that damping in one form or another exists at various points in the chassis, sometimes by accident, sometimes by design, and sometimes because it is unavoidable. It might be added that shock absorbers, commonly considered mainly a means of controlling vertical body movements are in addition expected to fill in or balance out all the deficiencies of the other damping means. The author shows that "a vehicle with relatively soft springs requires less damping than a vehicle with relatively stiff springs." To carry this thought a little further it follows that the softly sprung vehicle is more sensitive to minor variations in damping, and control of the damping becomes more imperative.

The selection of a shock absorber is thus not at all simple. The theoretical requirements are modified by the inherent damping in the particular suspension. Leaf springs provide damping not found in coil springs. The friction in shackles and in control arm bearings varies with the design of the parts and with the care used in as-

concluded on page 411

Development of REFERENCE F

by DONALD B. BROOKS*

Leader CFR-AFD Aircraft Fuel Knock Rating Scale Group and CFR Exchange Groups

MANY years ago the problem of knock rating had been solved. A single engine was widely accepted as standard, and with it a reference scale capable of indefinite extension. The Ricardo engine and the HUCR (highest useful compression ratio)[1] scale[2] appeared to comprise the Utopia of knock rating. All fuels then known could be rated in terms of this scale, and it seemed likely that any new fuel could be so rated, at least with minor modifications to the engine.

That Utopia was lost may be ascribed partly to man's preference for concrete physical reality. A compression ratio is about as tangible as a shadow. When a fuel was stated to have an HUCR of 5.3, that was just an abstract number, but a reference fuel was a substance which could be carried around in a can. Soon Utopia gave way to the Babel of individual reference fuels, tangible realities which were meaningless except to the possessor. This confusion was the worse confounded by the use of engines varying from small prime movers for farm lighting units to complete truck engines and single cylinders of aircraft engines.[3, 4, 5] The test results of each laboratory were practically unintelligible to other laboratories. Out of this welter of strange tongues a longing for uniformity gradually arose – uniformity of test engine and especially of reference fuels.

■ Development of Octane-Number Scale

As with many improvements in the art of estimating the quality of engine fuels, the concept of the octane-number scale was originated by a member of the Cooperative Fuel Research Steering Committee. A little less than 20 years ago, the *previously unknown* iso-octane, 2,2,4-trimethylpentane, was prepared under the supervision of Dr. Graham Edgar[6, 7] by J. W. McKinney. Dr. Edgar observed that it knocked less than any gasoline then available. He also obtained normal heptane, and noted that it knocked more than any gasoline then known, an observation made earlier by Ricardo. Engine tests showed that all gasolines then on the market could be matched with respect to knock by blends containing 40% to 60% of 2,2,4-trimethylpentane, or iso-octane, in normal heptane. Dr. Edgar suggested the use of these hydrocarbons as primary standards for knock rating. He proposed that the specification for U. S. motor gasoline be revised to include the requirement that the knocking tendency of this fuel be equal to that of a blend containing not less than 45% of iso-octane in normal heptane. He added that fuels equaling a 55% blend might well be reserved "for use in high-compression engines." Thus was the octane-number scale conceived.

For several years thereafter, the problems connected with standardization of a test engine and of the proposed reference fuels were debated in the Cooperative Fuel Research Steering Committee and its only offspring, the Detonation Subcommittee, and in the technical journals[8] as well. British resistance to the proposed heptane-octane scale, based largely on their use of heptane of variable purity fractionated from petroleum, was gradually overcome. On May 28, 1930, the Detonation Subcommittee, meeting at French Lick, Ind., approved the octane-number scale.

This, however, was not the octane-number scale which is now familiar. In the scale as first adopted, the octane number of a fuel was defined as the number of parts of 2,2,4-trimethylpentane in 10 parts of normal heptane, which was required to match the knock of the test fuel. On this scale, the F-3 octane number of present military aviation gasoline (Grade 100/130) would be infinity, and its F-4 rating would baffle Einstein. This particular shortcoming was soon recognized, and on Sept. 15, 1930, the Detonation Subcommittee reconsidered its earlier action and adopted the present scale. Approval by the Cooperative Fuel Research Steering Committee followed on the same day. In this scale, the percentage of iso-octane in blend with normal heptane which, in the standard engine, gives knock equal to that of a test fuel is called the octane number of the test fuel.

A proposal was made that the scale could be extended by adopting a value of 200 for some pure compound, such as alcohol or benzene, the knock rating of which was above that of iso-octane. The advent of the CFR Motor Method shortly thereafter hastened the demise of this proposal by rating alcohol at 91 octane number, and by showing that the rating of benzene also was dependent on test conditions. This was a blessing in disguise, as a scale so ex-

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[1] See Appendix for list of abbreviations used in paper.

[2] See Proceedings of Institution of Automobile Engineers, Vol. 18, Part 1, 1923-1924, Report of Empire Motor Fuels Committee, Section II, pp. 51-140: "Influence of Various Fuels on Performance of Internal-Combustion Engines." See also *Automobile Engineer*, Vol. 11, February, March, April, 1921, pp. 51-54, 92-97, 130-133; "Influence of Various Fuels on Performance of Internal-Combustion Engines," by H. R. Ricardo.

[3] See SAE Transactions, Vol. 22, Part I, 1927, pp. 24-33 + (disc.) 33-39: "Methods of Measuring Antiknock Value of Fuels," by H. K. Cummings.

[4] See SAE Transactions, Vol. 23, 1928, pp. 53-62: "Results of Two Recent Detonation Surveys," by H. K. Cummings.

[5] See *SAE Journal*, Vol. 25, July, 1929, pp. 80, 85: "Summary of Knock Test Methods."

[6] See *Industrial & Engineering Chemistry*, Vol. 19, January, 1927, pp 145.-146: "Measurement of Knock Characteristics of Gasoline in Terms of Standard Fuel," by G. Edgar.

[7] See *SAE Journal*, Vol. 20, February, 1927, pp. 245-246: "Detonation Specifications for Automotive Fuels," by G. Edgar.

[8] See 1946 CFR Handbook, pp. 282-285, for bibliography.

FUEL SCALES for KNOCK RATING

Report of Coordinating
Fuel Research Committee
of Coordinating Research
Council, Inc.

tended would have an intolerable discontinuity at 100. It also constituted a basic lesson which should have been, but was not, kept in mind in some later attempts at extension of the knock rating scale.

Nature of Octane-Number Scale

Standardization of technique soon resulted in a material increase in the precision of knock rating. The clearer insight gained thereby into the nature of the octane-number scale was disquieting. In the words of the late Harry L. Horning, the scale was a rubber yardstick. Before long it became apparent that the graduations on this rubber yardstick were far from uniform. Thus, the response of fuels to given additions of antiknock agents, such as tetraethyl lead, benzene, alcohol, and aniline, or to the proknock ozone, diminished as the octane number of the clear fuel increased. This is illustrated in Fig. 1, which shows the relative additions of exhaust gas required to increase the knock rating of a series of fuels by an equal amount. The relation of octane number to certain engine variables, illustrated in Fig. 2, shows a similar tendency. One way of expressing this effect is to say that the "size" of the octane unit increases as we go up the scale.

In 1933 it was pointed out in a memorandum[9] to the Detonation Subcommittee that this increase in the size of the octane unit was so rapid as to make the unit become infinitely large if the scale were extended to about 116. In other words, there is a limiting value to the octane-number scale, just as −273.16 C is the limit of the centigrade temperature scale.

Subsequent work has confirmed this conclusion in principle, although the more precise data now available place the end of the scale above 120 octane number by all present test methods. This can be deduced in any of several ways. The guide curves for setting compression ratio by the CFR Research and Motor Methods, if extended, would reach zero clearance volume, or infinite compression ratio, just above 120 octane number. A plot of the reciprocal of knock-limited indicated mean effective pressure (imep) against octane number is a straight line[10] which would reach zero (infinite imep) at 125-128 octane number for each of a number of supercharged engines. The extrapolated calibration curves of primary reference fuel blends having different constant additions of tetraethyl lead converge in this region. Even the proknock effect of ozone[11] appeared to reach zero near 120 octane number. It thus has been amply demonstrated that the octane-number scale is not uniform in terms of most other variables, and that it does not extend indefinitely.

The CFR Motor Method, developed in 1932 in response to the observation that road ratings of fuels did not agree with the Research octane numbers thereof, also introduced quantitative concepts of engine severity and of fuel sensitivity. Since the ratings of fuels vary from one engine to another, it follows that the knocking tendencies of some fuels must be more sensitive to operating conditions than are those of other fuels. This is termed fuel sensitivity, which is defined quantitatively as the difference between the Research and the Motor Method knock ratings of the fuel.[12] Thus we see that in addition to being nonuniform, the scale of knock rating will be warped to a greater or lesser degree as it is applied to engines of different severity.

PRACTICALLY all of the development of the art of knock rating has occurred during the past 25 years. The octane-number scale, an outstanding development 15 years ago, was fairly satisfactory until fuels rating above iso-octane became available.

Several means of extending this scale have gained limited acceptance, but experimental fuels have now exceeded the range of these systems.

A CFR group working on the problem has proposed a "triptane number" scale and an interchangeable detonation index, both extensible to the upper limit of antiknock quality.

Research now in progress may be expected to provide information on which adoption of this scale can be based.

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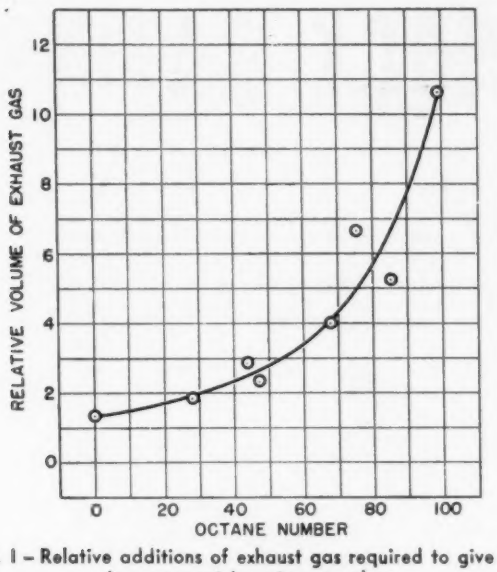
[9] See "Basis for Extension of Octane-Number Scale," by D. B. Brooks, unpublished memorandum to CFR Detonation Subcommittee, Oct. 21, 1933.

[10] See Journal of Aeronautical Sciences, Vol. 5, October, 1938, pp. 463-479: "Aircraft Fuels," by S. D. Heron and H. A. Beatty.

[11] See Journal of Institution of Petroleum Technologists, Vol. 19, October, 1933, pp. 835-844: "Ozone, Knock-Inducer Extraordinary," by D. B. Brooks.

[12] See *SAE Journal* (Transactions), Vol. 32, May, 1933, pp. 187-189: "Additional Opinions Concerning Antiknock Research—Road-Test Correlation," by D. B. Brooks.



■ **Fig. 1 – Relative additions of exhaust gas required to give equal improvement in octane number**

Given improvement in apparent rating of fuel requires greater volumetric addition of exhaust gas the higher the rating of fuel. Tests were made by CFR Motor Method

It should be noted that the term "engine" as used in this sense also includes the pertinent test conditions. The same physical engine, operated at a different speed, or with a different intake or coolant temperature, ignition, or valve timing, and so on, has a different severity, and thus constitutes a different "engine."

■ Extensions of Octane-Number Scale

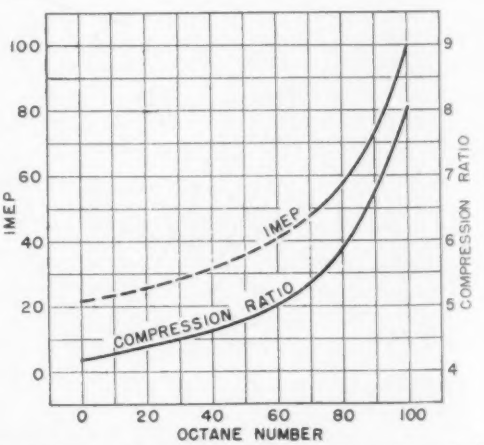
Improvement in the quality of aviation fuels and the development of test methods adapted to rating fuels of over 100 octane number led to increasing interest in extension of the knock rating scale. A number of methods was subsequently proposed, and some gained limited acceptance. Chief among these was the use of leaded blends of normal heptane and iso-octane, calibrated below 100 octane number and extended by assuming linearity or uniform curvature above that point. Such extrapolations, however, were uncertain because any small error in the calibration points was magnified by the extension, particularly where the calibration was nonlinear. A painstaking analysis showed that even the averaged data obtained by a group of laboratories in a planned series of calibration tests were not of a precision sufficient to warrant extension to 100% of the leaded iso-octane constituent. Needless to say, the extensions obtained by different test procedures were quite divergent.

Attempts were also made to extend the knock rating scale by the use of some engine variable in place of octane number. The performance-number scale, adopted by the military air services, has gained the widest acceptance of any such method. Developed initially as an average of the relative knock-limited output of a number of engines, from laboratory single-cylinder to full-scale multicylinder engines, operating on iso-octane clear and with additions of tetraethyl lead, this scale was subsequently extended to fuels below 100 octane number, as shown by Fig. 3. The term "performance number" is possibly an unfortunate one, as it has led many to believe that in some remarkable fashion, the performance number of a fuel exactly foretells its performance in any and all engines. This is not true, just as it is not true that a CFR Motor Method octane number of 70 means that the fuel will rate 70 octane number in all automobiles. The CFR Motor Method test conditions were chosen to rate fuels in the order in which they were *rated on the average* by a number of automobiles. The performance-number scale similarly gives the performance of fuels *on the average*, relative to iso-octane as 100.

More recently, the Subcommittee on Blending Octane Numbers of the Aviation Gasoline Advisory Committee has developed empirical extensions of the knock rating scale[13] for the CFR Aviation (F-3) and the CFR Supercharge (F-4) Methods. The extension for the F-3 Method is based on the octane-number scale, while that for the F-4 Method is an index number, having a value of 100 for iso-octane, and rather closely approximating the AN performance-number scale above 100. Both of these AGAC scales were developed to facilitate fuel blending computations, and are well suited to this purpose. The accuracy of estimating the knock rating of a blend containing several components of known individual ratings is stated to be about one octane number. However, it can be seen from a cross-plot of these scales (Fig. 4) that these extensions are less than ideal. Discontinuity does not occur in either *test method* at 100 octane number, but does occur in the *nature* of the *reference fuels*. The discontinuity at this point on Fig. 4, therefore, is attributable to faults in one or both of the extensions.

■ Work on Extension of Knock Rating Scale

A little over two years ago, the CFR Aviation Fuels Division organized a group on the Aircraft Fuel Knock Rating Scale. This group, hereinafter designated as the AFKRS Group, was assigned the dual problem of specifying reference fuels and a generally acceptable scale for the knock rating of aviation gasolines. As it was felt that the solution of the reference fuel portion of the assignment would have considerable bearing on the scale selected, the



■ **Fig. 2 – Relation of knock-limited compression ratio or indicated mean effective pressure to octane number**

These curves, as well as that of Fig. 1, suggest that the "size" of an octane unit increases toward the upper end of this scale

[13] See Report No. 5, Subcommittee on Blending Octane Numbers of Aviation Gasoline Advisory Committee, June 1, 1944.

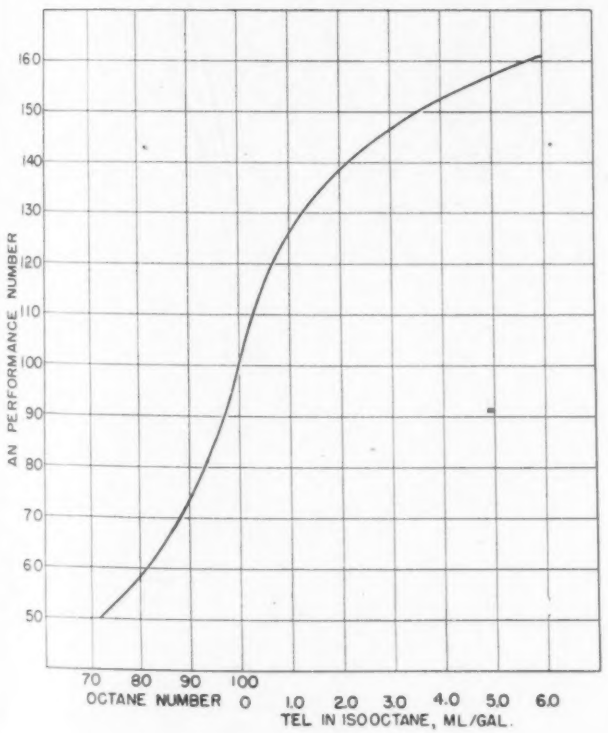
activities of the group were first directed toward the study of possible reference fuels.

Early experimental work of this group was devoted to a study of two sets of secondary reference fuels. One, designated LSM, was composed of technical iso-octane (S reference fuel) and a gasoline (M) of about 20 octane number, both containing 4.6 ml of tetraethyl lead per gal. The other, designated the LSMT scale, had in addition to the above, a constant toluene content of 10%, which conceivably might improve the precision of rating by making the sensitivity of the reference fuels comparable to that of the test fuels. As shown in Fig. 5, when blends of leaded alkylate and base stocks were rated by the F-3 Method against either set of reference fuels, a substantially linear relation resulted between rating and composition.[8] The shortcomings of the present reference fuel scale are well illustrated by the curve at the bottom of this figure. Similar charts for ratings by the F-4 Method are shown in Fig. 6.

From data similar to those shown in these figures, it was concluded that the LSMT scale was of sufficient promise to warrant further study.

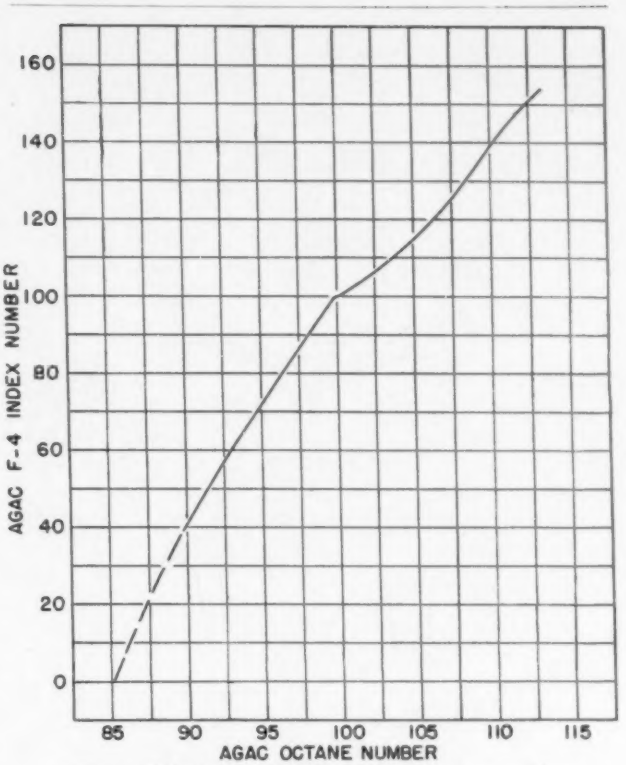
An inquiry into possible sources of supply of compounds suitable as components of a primary reference scale extending above leaded iso-octane showed that of the highly antiknock materials otherwise suitable, only triptane (2,2,3-trimethylbutane) could be made available in sufficient quantity. It was, therefore, decided to examine the possibilities of leaded blends of triptane and normal heptane (hereinafter designated LTrH) as primary reference fuels.

[14] See CRC report of CFR-AFD Aircraft Fuel Knock Scale Group on Tests of LSMT and Leaded Triptane-Heptane Blends, May 7, 1945.



■ **Fig. 3 – AN performance-number scale**

This scale, adopted by Army and Navy, is intended to relate fuel rating to average knock-limited engine performance



■ **Fig. 4 – Relation of AGAC index and octane numbers**

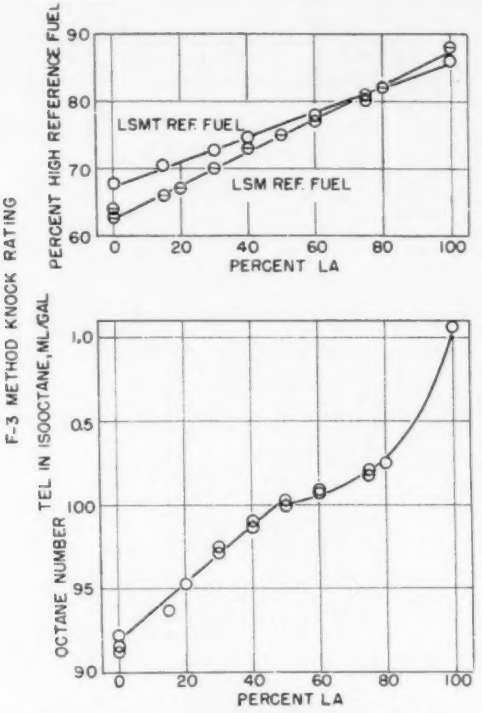
Discontinuity at 100 is evidence of imperfection in one or both of these scales, which were developed by the Aviation Gasoline Advisory Committee

A program was, accordingly, prepared for the intercomparison of LSMT, LTrH, and the present reference fuels.

The first part of this program called for the F-3 rating of 14 blends, containing from 48% to 100% LST, against the present secondary reference fuels, and the rating of iso-octane in terms of LSMT. Full mixture curves were to be run by the F-4 Method on 8 LSMT blends from 50% to 100%, and these were to be bracketed by present reference fuels. The study of the LTrH primary reference scale involved the intercomparison of LSMT and LTrH blends, and their comparison with iso-octane, by both F-3 and F-4 Methods. The six laboratories of the AFKRS Group participating actively in this program were: California Research Corp.; Intava Laboratories, Ltd.; National Bureau of Standards; Standard Oil Development Co.; Standard Oil Co. (Ind.); and Socony-Vacuum Oil Co.

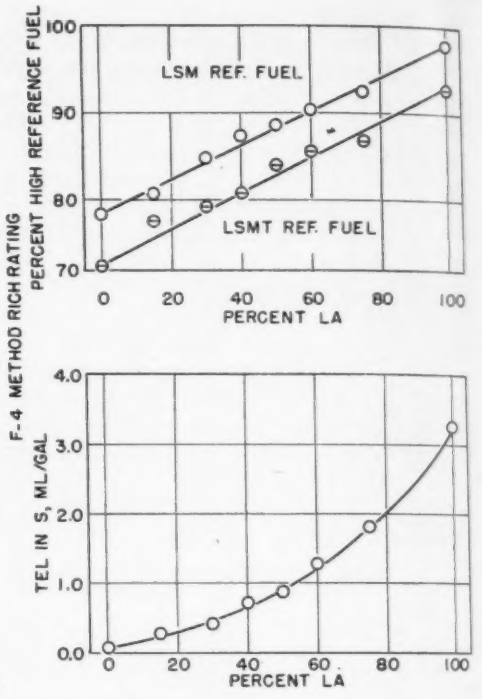
In spite of the fact that the LTrH blends, and also in some cases the LSMT blends, were prepared individually by the several laboratories, the agreement of the test data was very good.[14] In the F-3 rating of LST, for example, the results from five of the six laboratories fell within a range of 1.4 performance number.

The faired relations found between the various reference-fuel scales by the F-3 Method are shown by the solid lines in Fig. 7, the Army-Navy performance-number scale being plotted thereon as a dashed line. (A subsequent calibration by selected laboratories of the CFR-AFD Exchange Group, on a batch of LST and LMT fuels, confirmed the curves in this figure, and resulted only in straightening the F/C and octane-number lines.) In terms of the leaded triptane-heptane blends, the F in C, the LSMT, and the octane-number scales are all nearly linear, as is the performance-



■ **Fig. 5 – Ratings of leaded alkylate-base blends by CFR Aviation (F-3) Method**

Ratings are expressed in terms of octane number and of tetraethyl lead in iso-octane in lower figure; above, they are given in terms of leaded S and M (LSM) and leaded S and M with 10% toluene (LSMT) scales



■ **Fig. 6 – Ratings of leaded alkylate-base blends by CFR Supercharge (F-4) Method**

Ratings are expressed in terms of tetraethyl lead in technical iso-octane (S reference fuel) in lower figure; above, they are given in terms of leaded S and M (LSM), and leaded S and M with toluene (LSMT) scales

number scale over the range 60% to 90% LTrH. The experimental secondary scale, LSMT, is seen to cover the range of present reference-fuel scales by this method, and the contemplated primary reference-fuel scale, LTrH, is seen to extend well beyond the present scale.

Fig. 8 is typical of the F-4 Method rating curves obtained by members of the AFKRS Group in the investigations of the LSMT and LTrH scales. From the averaged data, the interrelations of the various scales were found to be as shown in Fig. 9. It is noteworthy that by this method, all scales composed of two reference fuels having zero or equal content of tetraethyl lead are related by straight lines. It is also of interest that only 81% of leaded triptane in heptane is required to equal the upper limit of the present scale, iso-octane plus 6 ml tetraethyl lead per gal. The leaded triptane-heptane scale thus appears to be more than adequate for any probable extension of the F-4 Method.

■ Relation of Imep to Fuel Composition

As mentioned earlier, it has been observed that a plot of the reciprocal of knock-limited indicated mean effective pressure (imep) against fuel composition is generally linear, when determined in supercharged engines. In Fig. 10, the reciprocal imep at the F-4 rich peak is plotted against the LTrH scale. It is obvious that this relation is linear over the relatively large range of 25% to 85% LTrH. Extrapolation, justified by the range and linearity of the data, shows that the reciprocal imep would become zero at 109.5% LTrH. This point is the *absolute* upper limit of knock scales by the F-4 Method, or by any method of equal severity, as with a fuel of this rating, trace knock would occur only at infinite supercharge pressure. The numerical value of this endpoint is not fixed exactly by the few data shown here, but the occurrence of an endpoint is beyond dispute.

Fig. 10 opens a vista of interesting and useful deductions. In the first place, the fact that the infinity of the F-4 knock scale is but 9.5% beyond the physically realizable 100% LTrH, and that determinations may be made to as low as 25% LTrH, means that the error in extrapolating to the infinity of the scale is small. Here, for the first time, is a continuous scale which can cover the range of all present and future aircraft fuels. Should it become necessary to rate fuels above 100% LTrH by a test method of severity comparable to that of the F-4 Method, a secondary scale having a component above 100% LTrH can be calibrated up to that point, and used at higher levels with complete confidence in the results.

Another deduction from Fig. 10 is that in evaluating knock ratings by interpolating between reference fuels, the interpolation should be based on the reciprocals of the indicated mean effective pressures, not on the direct values, which are not linear with reference fuel composition. For example, if 60% and 70% LTrH are used as bracketing reference fuels, the error so arising from interpolation between the direct values of the indicated mean effective pressures would be greatest near the center of the bracket, and would amount to 0.6% LTrH. Over more than half the bracket range the error would exceed 0.4% LTrH. This error varies in magnitude over the LTrH scale, increases with the square of the bracket size, and is always

negative in sign within the bracket, and positive outside the bracket.[15] Any extrapolation leads to large error. For example, a fuel equal to 75% LTrH, if thus rated from a 60% to 70% LTrH bracket, would be evaluated 2.2% LTrH too high.

However, when interpolation is based on the reciprocal imep, no such curvature errors are present, and the size of the bracket is without effect on the precision of rating. Using data from one laboratory, which covered the range 25% to 85% LTrH, and interpolating between these limits to determine the ratings of their intermediate blends from the reciprocals of the reported imep values, yields results with a standard deviation from the nominal values of 0.56% LTrH. Note that this interpolation covers the range equivalent to about 40 octane units with an *actual* – not a *calculated* – precision of about one-third of an octane unit!

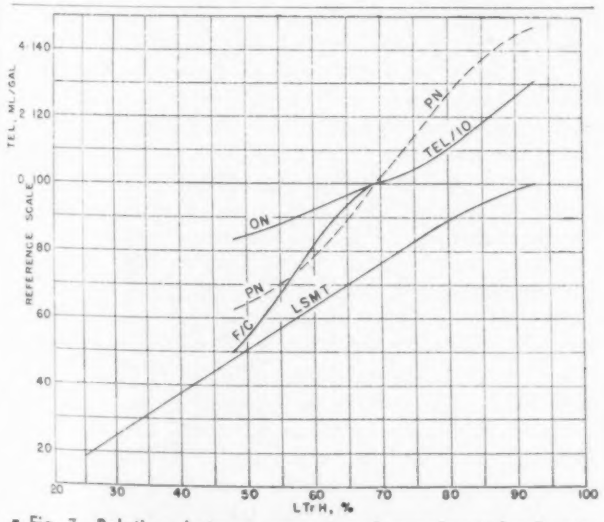
While the routine use of reciprocals would involve additional computation, this can be avoided, and the entire process of obtaining an F-4 rating can be simplified, with an accompanying increase in precision, by a suitable revision of the standard curve sheet. This revision is accomplished by so warping the imep framework as to convert reference-fuel response curves into horizontal straight lines spaced equally. On such a plot, test-fuel response curves are approximately linear and nearly horizontal, and the rating is obtained by direct interpolation between the reference-fuel response lines.

■ Detonation Index

Also from Fig. 10, the ratio of the knock-limited indicated mean effective pressure of any LTrH blend to that of iso-octane can be found readily. As iso-octane is rated at 65.5% LTrH, the value of 1000/imep for iso-octane, divided by the corresponding value for the LTrH blend in question gives the desired ratio. When multiplied by 100, this ratio gives a detonation index which indicates the relative performance of the fuels in the test engine. This detonation index is continuous and logically extensible to any desired limit. It can be determined physically on the test engine. It is not a function of any given reference fuel *scale*, as it is based on iso-octane alone, but it can be correlated with any scale.

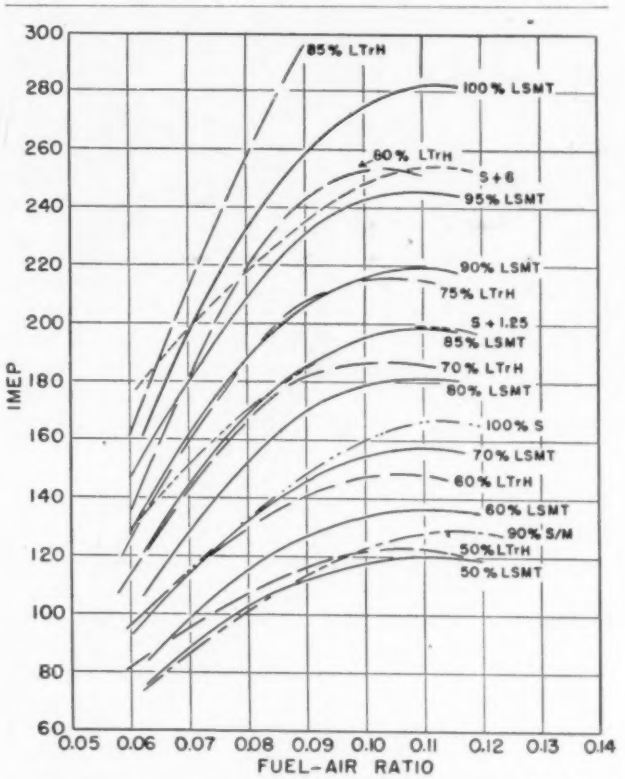
Fig. 11 shows the AN performance number and the detonation index plotted in relation to the LTrH scale. As the detonation index is the actual imep – or power – ratio of the LTrH blends to iso-octane in the F-4 test engine, it is apparent that the AN performance number does not correctly indicate performance in this engine, in the range above 100. Originally, a rough average of the performance ratios of an assortment of engines operated on fuels from clear iso-octane to iso-octane plus 6 ml tetraethyl lead per gal, the AN performance number above 100 is now defined by a formula relating it to tetraethyl lead in iso-octane. This formula is usable only to iso-octane plus 6 ml tetraethyl lead per gal, which is equivalent to 161 performance number. When the AN performance-number scale was extended below 100, a totally different type of formula was chosen. This new formula is of the type used in relating the detonation index to the LTrH scale. This fact and the choice, implicit in the formula, of 128 as the end of the octane-number scale account for the excellent agreement of the detonation index and the AN performance-number scales below 100. By the present work, the end of the octane-number scale is set at 127.4.

[15] See "Effect of Bracket Size on Precision of Detonation Rating," by D. B. Brooks, Appendix C, minutes of meeting of CFR Detonation Subcommittee, April 14, 1931.



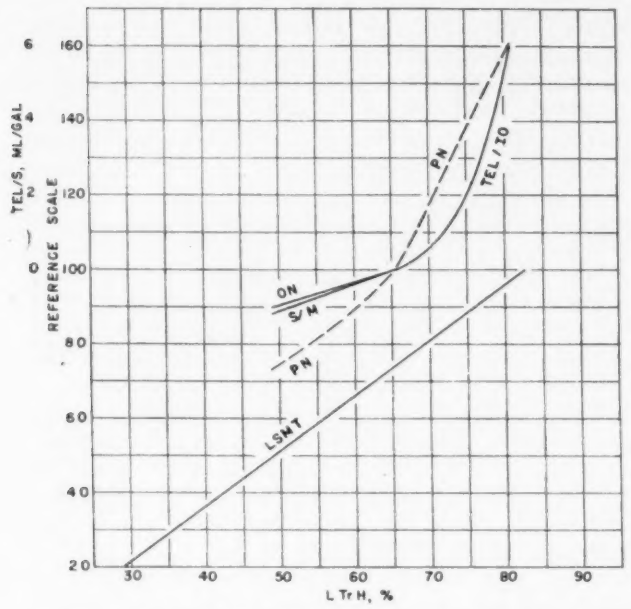
■ Fig. 7 – Relations between present and experimental reference fuels by CFR Aviation (F-3) Method

Scales shown are present secondary reference scale of fuel F in fuel C (F/C), octane number (ON), tetraethyl lead in iso-octane (TEL/10), leaded S and M with 10% toluene (LSMT), and the Army-Navy performance-number scale (PN) in comparison with leaded triptane-heptane (LTrH)



■ Fig. 8 – Typical rating curves by CFR Supercharge (F-4) Method

Curves of knock-limited indicated mean effective pressure (IMEP) against fuel-air ratio are shown for blends of reference fuels S and M (S/M), for tetraethyl lead in S (for example, S + 6), and for various concentrations of leaded S with 10% toluene in leaded M with 10% toluene (LSMT), and leaded triptane-heptane (LTrH)



■ **Fig. 9 – Relations between present and experimental reference fuels by CFR Supercharge (F-4) Method**

Scales shown are fuel S in fuel M (S/M), octane number (ON), tetraethyl lead in iso-octane (TEL/10), leaded S and M with 10% toluene (LSMT), and the Army-Navy performance-number (PN) scale in comparison with leaded triptane-heptane (LTrH)

The fact that actual performance ratios are not linearly related to fuel composition, as is shown in Fig. 11, means that such scales are inherently unsuited to fuel blend computations. On the other hand, all of the reference-fuel scales for the F-4 Method, except those having variable lead content, are linearly related, as shown in Fig. 9. Furthermore, the knock ratings of most blends, such as alkylate and base, are approximately straight-line functions of composition when the rating is expressed in terms of reference fuel scales such as those under consideration, provided, of course, that the lead contents of the components are equal. There are exceptions to this generalization, chiefly in cases where the blend components are of diverse types of hydrocarbons, but such exceptions do not follow any rule applicable also to the normal blends. Hence it appears that the LTrH scale, or "triptane number" as it might well be called, is well suited for use in blending computations.

In engine work, performance is of more concern than fuel composition. Experimental fuels now in use have surpassed the useful range of the AN performance-number scale as defined, and performance numbers are assigned to these fuels on the basis of the ratio of the knock-limited imep of the experimental fuel to that of iso-octane, in the test engine. Such performance numbers are identical both in principle and in practice with the proposed detonation index. As the detonation index has thus been accepted in practice, it would appear desirable to abandon the AN performance-number scale altogether in favor of the detonation index, thus having a scale which is factual, illimitably extensible, and rigorously related to the reference-fuel scale used in blending computations. The amount of change involved in transforming AN performance number to detonation index in the range of present production fuels can be seen from Fig. 11.

It should be noted that the detonation index as developed for the F-4 Method can not be applied to the F-3 Method. In the latter method, the compression ratio and consequently the severity is increased with rating level. The reciprocal relationship of knock-limited indicated mean effective pressure and reference fuel composition does not hold in such a case. However, as shown in Fig. 12, the F-3 rating is closely related to the F-4 lean minimum rating.[18] The latter was abandoned recently because of poor reproducibility, but a lean-minimum method may be reinstated in place of the F-3 rating when the improved supercharge method now under development is completed. As the reciprocal relationship was found to hold also for F-4 lean minimum ratings, it may be expected to hold in the case of the new method, and a suitable detonation index can, therefore, be developed for this rating.

■ Discussion of Proposed Scales

Both of the scales under study have the following advantages over present scales: (a) continuity over the range at present of interest, thus simplifying the treatment of test data, and facilitating blending computations; (b) constancy of tetraethyl lead content, which, together with the fact that this content approximates that of most aviation fuels, should result in a reduction of the error of rating, and should expedite rating by eliminating "lead hangover," which often occurs when changing from the test fuel to reference fuels either clear or of different lead content; and (c) sensitivity approximating that of conventional aviation fuels, which should reduce rating errors arising from this cause.

It has been argued that the lead content of the proposed reference fuels is disadvantageous from the standpoint of servicing required on the knock test engines. This argument loses much of its force when it is recalled that the effects of tetraethyl lead are more serious at high output, and that under these conditions the present reference fuel, iso-octane, requires a comparable or greater addition of tetraethyl lead to match fuels capable of high output.

It would be contrary to all precedent, at least in the field of knock rating, if the triptane scale were not accompanied by some disadvantages. The objection most generally raised is the high sensitivity of triptane. The knock rating of clear triptane by rating methods currently in use varies from only slightly higher than that of iso-octane to far above iso-octane plus 6 ml tetraethyl lead per gal. That the addition of tetraethyl lead to triptane does not reduce this sensitivity greatly can be inferred from the fact that iso-octane plus 3 ml tetraethyl lead per gal is equal to 77% LTrH by the mild F-4 Method (Fig. 9), while by the severe F-3 method it equals 92% LTrH (Fig. 7). Two rejoinders can be made to this objection.

In the first place, iso-octane is less sensitive to changes in engine severity level than almost any fuel of equal rating. It is much less sensitive than any commercial aviation fuel, as is indicated by the disparity of the two grade numbers assigned to the latter, 100/130 for example. This lack of sensitivity is by no means an unmixed blessing. In rating fuels against iso-octane, any vagary of the engine severity will affect the knock of the test fuel to a much greater extent than that of the iso-octane reference blend, and will thereby alter the rating of the test fuel. The

[18] See "Precision of Rating Aviation Fuels," by D. B. Brooks, CRC report, Jan. 7, 1945.

precision of fuel rating, therefore, may be expected to be poorer with iso-octane than with a suitably sensitive fuel. The behavior of triptane in this respect can be indicated by the fact that grade 100/130 fuel, which by the F-3 Method is equal to iso-octane, and which is equal to the latter plus 1.25 ml tetraethyl lead by the F-4 Method, would rate 69 and 73 triptane number respectively. Thus, at this level blends of triptane and heptane are still slightly less sensitive than this well-known military aviation fuel.

Secondly, from test data which can not be disclosed in full at present, it is beginning to appear that high sensitivity is an inevitable corollary of high knock rating. Should this prove to be correct – and there is but little doubt that it will – then it follows that any physical scale above iso-octane will involve a sensitive fuel, and also that any aviation fuels of higher ratings than those now used will be of higher sensitivity than current fuels, and preferably should be rated against sensitive reference fuels.

With the completion of research now being carried on by the CFR-AFD Exchange Group on the calibration of a pure batch of triptane, and on the routine rating of monthly samples against the experimental secondary scale, LSMT, definite information will be available on the relative precision of rating with present and experimental scales. This information, together with that obtained earlier by the AFKRS Group, should provide an adequate basis for evaluating the relative merits of the present and the proposed scales. Should the triptane scale then be adopted for use with aviation fuels, it would be desirable to study its application to the motor fuel test methods. The advantages of having a single scale which would cover all test methods are too obvious to warrant discussion.

Conclusions

The past quarter of a century has witnessed practically all of the development of the art of knock rating. The pioneer Ricardo engine and reference scale gave way to a miscellany of engines and scales, from which emerged the CFR engine and the octane-number scale. With the advent of fuels rating above 100 octane number, several systems of extending the octane-number scale were proposed. Of these the principal survivors are the AN performance number and the AGAC extended octane and index numbers. Experimental fuels have now exceeded in knock rating the range of these systems.

With increasing study of the problem it has become apparent that a single rating scale will not satisfy the needs of all concerned. A scale is needed in terms of which fuel blending computations can be made readily, and an interchangeable scale is needed by means of which relative engine performance can be expressed. While it is true that such a performance scale can be exact only for the engine on which it was developed, it is likely that with increasing understanding of the problem we shall be able to estimate accurately the performance of the fuel in any given engine from ratings of the fuel under mild and severe test conditions.

The exploratory work of the AFKRS Group of the CFR Aviation Fuels Division leads to the conclusions that the leaded triptane-heptane scale appears to have much promise; that this scale, which is expressible either as triptane number or detonation index, being extensible with accuracy to the upper limit of antiknock quality, is of adequate coverage; and that it is definitely superior in important respects to present reference fuel scales. Research now in progress may be expected to provide information on which adoption of this scale can be based.

Appendix

List of Abbreviations Used in Text

AFD = Aviation Fuels Division of the CFR

AFKRS = Aircraft Fuel Knock Rating Scale Group of the CFR Aviation Fuels Division

AGAC = Aviation Gasoline Advisory Committee

AN = Army-Navy

C = Standard secondary reference fuel of roughly 70 octane number

CFR = Cooperative Fuel Research (now the Coordinat-

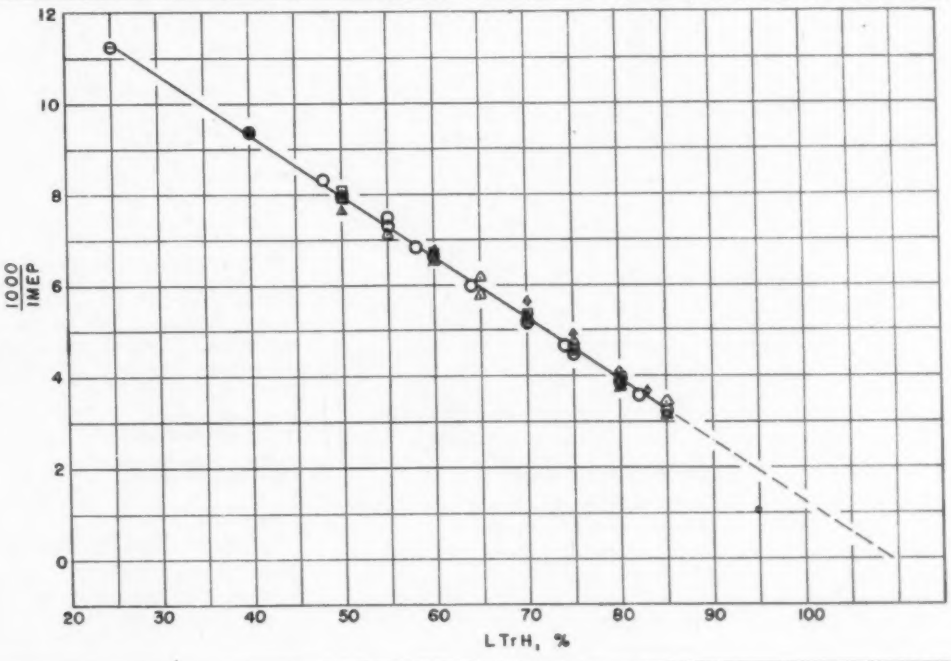
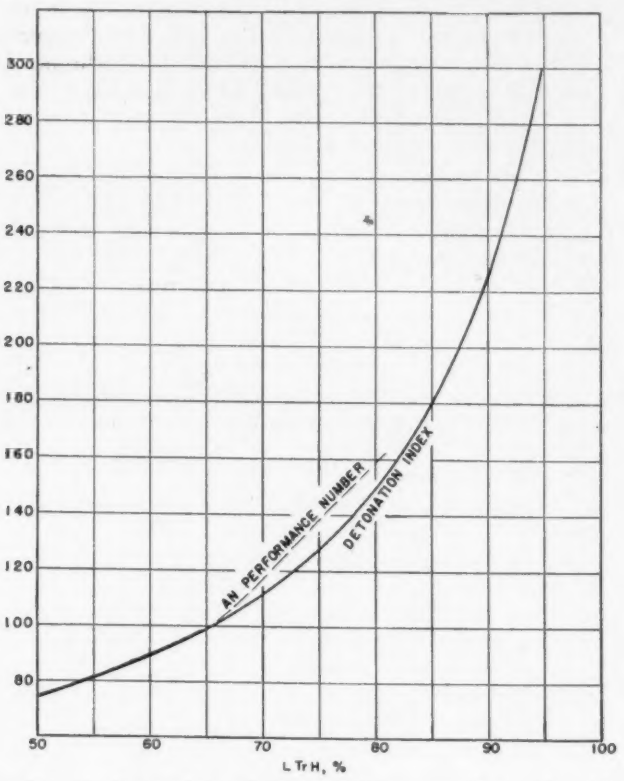


Fig. 10 – Variation of reciprocal knock-limited indicated mean effective pressure with reference fuel composition. Tests by CFR Supercharge (F-4) Method show that a linear relation exists between knock-limited indicated mean effective pressure (IMEP) and leaded triptane-heptane (LTrH) scale



■ **Fig. 11 – Comparison of AN performance-number scale with detonation index**

Detonation index is actual ratio of knock-limited power of leaded triptane-heptane blends (LTrH) to that of iso-octane taken as 100. Army-Navy performance number, an average ratio for a number of engines, does not correctly indicate power ratio above 100 in CFR Supercharge (F-4) Method engine

ing Fuel Research)

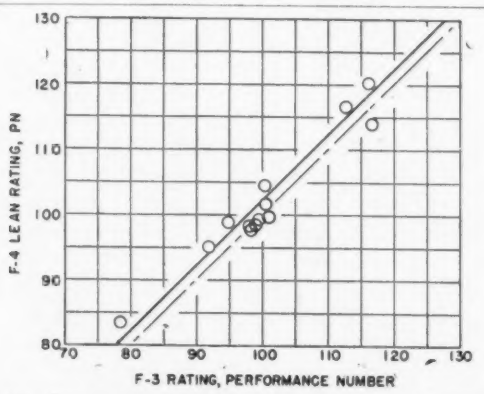
F = Secondary reference fuel composed of technical iso-octane

F-1 = CFR Research Method

F-2 = CFR Motor Method (ASTM Method D 357)

F-3 = CFR Aviation Method (ASTM Method D 614)

F-4 = CFR Supercharge Method



■ **Fig. 12 – Comparison of CFR Aviation (F-3) and CFR Supercharge (F-4) lean minimum ratings**

Ratings by these two methods, expressed in terms of the Army-Navy performance-number scale, agree closely for a number of typical aviation fuels. Solid line fits data, while broken line indicates exact equality. Difference is equivalent to about one-half octane unit.

Grade 100/130 = Military aviation fuel rating 100 octane number by the F-3 Method and 130 performance number by the F-4 Method

HUCR = Highest useful compression ratio, or ratio at which light knock occurs with ignition setting and mixture ratio adjusted for best power

LMT = Secondary reference fuel of M gasoline containing 10% toluene and 4.6 ml tetraethyl lead per gal

LSM = Blend of secondary reference fuels S and M containing 4.6 ml tetraethyl lead per gal

LSMT = Blend of secondary reference fuels S and M containing 10% toluene and 4.6 ml tetraethyl lead per gal

LST = Secondary reference fuel S containing 10% toluene and 4.6 ml tetraethyl lead per gal

LTrH = Blend of primary reference fuels triptane and heptane containing tetraethyl lead. In the original work, lead concentration was 4.6 ml per gal. Current blend contains 3.785 ml per gal, or one part per thousand

M = Secondary reference gasoline of about 20 octane number

S = Secondary reference fuel of technical iso-octane. Very similar to fuel F

DISCUSSION

Value of Performance-Number Scale For Military Personnel Pointed Out

—**S. D. HERON**

Ethyl Corp.

THE problem of control of engine severity is a difficult one in knock rating as regards both laboratory and full-scale aircraft engines. While all engine conditions, such as jacket temperature, cylinder temperature, and intake air temperature, can be held to standard values, variables affecting severity can occur and have significant results, particularly when rating a sensitive fuel in terms of the relatively insensitive iso-octane. Intake and exhaust valve condition can have serious effects upon engine severity and a leaking intake valve can be particularly objectionable. It is my personal view that the leaded triptane-heptane reference fuel scale is much the best which has been offered to date for aviation fuels, and I find it difficult to visualize a scale which is as good, let alone better. To the best of my knowledge the sensitivity of this scale varies considerably, depending upon the amount of triptane present. However, the sensitivity of any other possible scale (including heptane-octane or heptane-octane-lead) also appears to increase as the amount of high knock rating component is increased. Thus, any foreseeable scale will not produce the desirable effect of always having the test and reference fuels of something like equal sensitivity. With aviation fuels and any reference fuel scale the practice of requiring mild and severe engine ratings, such as F_3 and F_4, now required appears to be a very sound one and to provide a considerable measure of information concerning the test fuel no matter what reference fuel scale is used.

Dr. Brooks is somewhat skeptical of the performance-number scale. Whatever the demerits of this scale it has been of great value, particularly with military personnel, in promoting understanding of the importance of not using a fuel lower than the specified grade for any particular aircraft. With military personnel the statement that a fuel had a knock rating equal to iso-octane plus 1 cc lead meant almost nothing and for this reason the description of fuels in some scale such as performance number or detonation index, which bears an approximately linear relationship to engine performance, would appear to be desirable in the future.

Need Emphasized For Longer Scale

—**WHEELER G. LOVELL**

Research Laboratories Division, General Motors Corp.

THE facts underlying the use of reference fuel scales are quite simple. The first fact is that some fuels are better than others in respect to knock; and it would be nice to have some numerical scale for telling how much better. The second fact is that how much better a fuel is depends upon the engine in which the comparison is

made; and this fact is very unpleasant: We know of some fuels that may be alike in one engine and unlike in another by a quarter of the distance between the two ends of the octane-number scale.

Most of the troubles we have been through and are going through exist because, although the first fact that some fuels are better than others has been generally known for about 25 years, the second fact, that a fuel rating depends upon the engine, has been known for a much shorter time and perhaps has been widely recognized only after a reference fuel system was pretty well adopted.

As a result, we have all kinds of octane numbers and road octane numbers and performance numbers and detonation indexes, and discussions that are sometimes productive of more heat than light. But we can rightfully hold to one thing, namely, that all fuel ratings are based, irrespective of how they are computed after the actual comparison in an engine is made, upon a comparison of mixtures of n-heptane and iso-octane, or when the scale proved too short, of iso-octane with lead.

Dr. Brooks quite soundly proposes that, inasmuch as fuels have improved so that the octane-number yardstick is not long enough, we adopt a new yardstick at least to supplement the old one, based on leaded mixtures of heptane and triptane. This scale seems very logical and sound as far as we know now, and a lot of work is still going on. It certainly seems a long enough scale, because triptane is a good fuel, and the ratings in Dr. Brooks' paper only run up to about 85% triptane, possibly because no one had simultaneously enough nerve and enough engine to go higher.

Certain it is that we need this longer scale, because better fuels are bound to come, since we need better overall fuel economy for many applications; better engines may be expected to come with them, because without some changes in engines, there are no super fuels, as Dr. Brooks has pointed out.

Suggests Further Improvement In Author's Knock Rating Scale

—ROBERT V. KERLEY
Ethyl Corp.

DR. BROOKS has proposed that the present system of primary reference fuels be replaced by the leaded triptane-heptane system. It has been simultaneously proposed that the detonation index be adopted interchangeably to express relative engine performance. An analysis of these joint proposals is necessary before the full complications of such a dual system can be realized.

The need for a new system of reference fuels is indicated, and such a system will probably be adopted. Whether this system is the leaded triptane-heptane system proposed or is some variation of it is, in many respects, unimportant as compared to the method of application of the reference fuels. If a new system of primary reference fuels is to be adopted this new system should be independent of the system which it replaces. This cannot be true if the detonation index value for any blend of reference fuels, such as triptane in heptane, is dependent on the rating of that blend relative to iso-octane, since most of these blends then have an infinite number of detonation indexes as the fuel-air ratio or any other engine operating condition is changed.

To accept the detonation index as defined in the basic paper would effectively nullify any system of reference fuels, since the test fuel rating is then relative to iso-octane only and is completely independent of any bracketing reference fuels. Therefore, it may be concluded that Fig. 11, illustrating the correlation between the leaded triptane-heptane blends and detonation index in the F-4 test engine, is of no importance unless this correlation is applied in a manner which has not been indicated.

Some simplification would result if the value for expressing relative performance were redefined in terms of the proposed reference fuel. For example:

$$\text{Value expressing performance} = \frac{\text{Imep of test fuel}}{\text{Imep of 65.5\% LTrH}}$$

Such a ratio has the advantage that the new primary reference fuel is actually independent of the reference fuel it is replacing. However, in this instance there is no reference fuel *system*, since *a single blend is the reference fuel*, and bracketing of the test fuel is, by definition, not required. An obvious means of overcoming this defect is to define the value expressing performance in terms of the reference fuel blends. For example, the detonation indexes for the various leaded triptane-heptane blends might readily be defined by Fig. 11. Under this system the bracketing of the test fuels by blends of reference fuels is required to determine the blend of reference fuels equivalent to the test fuels, and reference to Fig. 11 is necessary to determine the detonation index, the performance number, or some such value for expressing relative performance.

It will be noted that the method of application finally suggested is identical to the *correct* method of applying the performance number when the heptane-iso-octane-lead scale of rating is used. The advantages of the proper application of the proposed leaded triptane-heptane reference fuel system are: that, since a constant lead level is used, it is possible to use ratings in terms of equivalent reference fuel blends for blending calculations; that it is possible by plotting inverse mean effective pressures to obtain an extrapolated rating for a test fuel above leaded triptane, and that the reference fuel system is continuous from a reasonably low value to a value high enough to cover the maximum ratings normally encountered even in experimental testing.

Need Shown for Clear Understanding Of Fuel Blending Characteristics

—HENRY C. BARNETT
NACA

FOR my discussion of Dr. Brooks' paper I have chosen to elaborate on a few of the points of interest in connection with the choice of a reference fuel rating scale. There are two facts that stand out in this paper: (1) that a clear understanding of fuel blending characteristics is necessary to the development of a rating scale, and (2) that the sensitivities of fuels to changes of engine severity make the selection of a foolproof rating scale a near impossibility. These two facts have been known for many years, but we have all been guilty at times of disregarding their importance.

With regard to the first point, fuel-blending characteristics, the first published use of the reciprocal blending relationship was made by Heron and Beatty in reference 10 of the subject paper. In this reference the authors found that by plotting the reciprocal of the imep against the composition of blends of S and M reference fuels a straight line was obtained. This relationship was found to be linear under three sets of operating conditions. It should be emphasized, however, that blends of S and M fuels are predominantly paraffinic and, as Dr. Brooks indicated, the relationship does not hold for diverse types of hydrocarbons.

I should like to go further into the restrictions on this relationship to list a few of the conclusions that are indicated by work performed at the NACA Cleveland laboratory. These are as follows:

1. The relation between reciprocal imep and composition holds for blends of paraffinic fuels tested in supercharged engines, provided the conditions of operation are not extremely severe.

2. Aromatic hydrocarbons do not follow this relationship even for mild operating conditions.

3. In cases where the relation does not hold it appears that the nonlinearity obtained may be due to lack of control over cylinder-wall temperatures—a factor of obvious importance in relation to F-4 engine.

4. If an additive is present in the two components to be blended, then the concentration of the additive must be the same in both components. This fact was mentioned by Dr. Brooks.

These restrictions are offered to allay any possible beliefs that the reciprocal blending relationship is a hard and fast rule. When the user of the relationship is aware of these limitations the method can be a useful tool from the standpoints of rating fuels (as Dr. Brooks has shown) and estimating ratings of blends.

My next point for discussion concerns the sensitivity of fuels to changes of engine operating severity. Here we have the most critical problem of all in the development of a suitable rating scale. Since fuels differ in performance as engine-operating conditions are changed the selection of any two reference fuels for a permanent rating scale will be hampered. Thus, as we all know, we can never expect the numerical rating of a fuel to be the same under all conditions and in all engines. As Dr. Brooks has stated, however, for fuels in the high-performance category the constancy of numerical rating may be more closely approached by use of a reference fuel as sensitive to engine conditions as triptane. This is true if our belief that high sensitivity and high knock rating go hand in hand is proved conclusively.

In conclusion, I should like to bring out one consideration that Dr. Brooks has not mentioned with regard to the proposed triptane scale. This consideration is related to the problem of sensitivity. One of our primary jobs in fuel research is to evaluate new compounds in terms of knock ratings obtained by our current rating methods, F-3 and F-4. It can safely be said that many such compounds will have ratings in excess of triptane leaded to 4 ml per gal under severe conditions, such as those imposed by the F-3 engine or by the F-4 engine at lean mixtures. Thus, where previously the fuel research worker had an F-3 yardstick up to a performance grade of 161 the use of triptane leaded to 4 ml per gal would impose a performance limit of about 152. This difficulty may be alleviated by the possibility of using the lean rating of the knock rating method now.

GERMAN

STUPENDOUS efforts were made in the basic research of guided missiles by the Germans. In all it has been estimated that one-third of the aerodynamics research in Germany was devoted to problems of these weapons.

Wind tunnels of undreamed-of speeds were under construction when the war was over. In fact a Mach number of 10 was not too great for the Germans to comprehend and strive for.

The various types of developments grouped under the expression guided missiles (a loose term as some aren't guided and some aren't missiles) are discussed here by Col. Putt.

THE AUTHOR: COL. D. L. PUTT, USAFF, has since August, 1945, been assigned as deputy commanding general, Intelligence (T-2), in charge of all technical intelligence activities of the Air Materiel Command at Wright Field. He joined the Air Forces as a flying cadet in 1928, and after completing his flying training was assigned to Wright Field. In January, 1945, he was made director of technical services of the ATSC in Europe. His service awards include the Legon of Merit for work on the B-29, Bronze Star with Oak-Leaf Cluster, and the Croix de Guerre.

THE subject of guided missiles involves many fields of science; namely: aerodynamics, supersonics, ordnance, ballistics, compressibility, electronics, telemetering, control, fuels, and others.

The science and development of guided missiles and their propulsive units were exploited to the greatest extent by German scientists prior to V-E Day.

Rockets as defensive weapons caused very little excitement in Germany until 1943, although a tremendous amount of experimentation had been accomplished by pure rocket enthusiasts since 1925. As the war progressed it became evident that Germany would eventually be on the defensive, and rocket weapons came into prominence to a greater and greater extent. These were the so-called "secret weapons" Hitler had promised his people. They were to have reversed the course of the war and given Germany a new and startling shortcut to victory. After watching the V-1 and V-2 firing trials at Blizna and Cracow, Poland, in April, 1944, Hitler is reported to have stated that German secret weapons were not the product of dreamers and that England and the whole world would soon feel their effect. It wasn't until allied technicians examined German developments in this field that we fully realized the tremendous achievements of German scientists, and how near they were to achieving the boasts of their leader.

The Germans were preparing rocket surprises for the whole world in general and England in particular, which would have, it is believed, changed the course of the war if the invasion had been postponed for so short a time as six months. Many of Germany's research laboratories and several large commercial firms concentrated on this field of endeavor. This tremendous effort resulted in 138 guided missiles and assorted devices, including their modifications. These were of types wholly unknown to laymen in the United States. At the outbreak of the war some of these were strictly "out of this world" – to use a current phrase. In addition, German scientists had developed other equipment of a type we had considered impracticable, such as the ram jet.

The stupendous effort in basic research expended by the Germans in the guided missile field was designed to cover the complete field of potentialities for such weapons. The losses incurred in Germany by heavy bomber raids can in no way be charged to lack of preliminary research on missiles. Weapons of this category were divided into the following classifications:

A. Ground to air.
B. Air to air.
C. Air to ground.
D. Ground to ground.
E. Underwater to underwater.
F. Underwater to ground.
G. Underwater to air.

Moreover, every known type of remote control and fusing means was being exploited. These included radio control, wire control, radar, continuous wave, acoustics, infrared, light beams, and magnetics.

Likewise, all methods of employing jet propulsion for subsonic and supersonic speeds were exploited.

In all, it was estimated that one-third of the aerodynamics research in Germany was devoted to problems of guided missiles. Wind tunnels of undreamed-of speeds were under construction. A Mach number of 10 was not too great for the Germans to comprehend and strive for.

In the matter of aerodynamics and propulsion, the Germans were sufficiently advanced to handle any program desired. In the field of control they appear to have waited too long to make the necessary tests to indicate the proper directions for detail development; also, the fact that the Germans were late in being able to arrive at the answers required to utilize nuclear energy in the warheads made the German missile program more one of nuisance and worry over what might come than one of actual military damage.

Some of the classes of German missiles based on intended use are:

A. The Beethoven (Fig. 1) was an air-to-ground missile. This composite aircraft consisted of an Me-109 fighter mounted on top of a Ju-88 bomber. The pilot rode in the cockpit of the Me-109, where he manipulated the controls of both craft. The nose of the Ju-88 was modified to contain an explosive. Upon reaching the target area the Ju-88 was released from the Me-109 in a dive which aimed it at the objective. This weapon was not used by the Germans to any extent; however, upon the termination of hostilities

[This paper was presented at a meeting of the Metropolitan Section of the SAE, New York City, March 7, 1946.]

DEVELOPMENTS in the FIELD of GUIDED MISSILES

by COL. D. L. PUTT
Deputy Commanding General for Intelligence (T-2)
Air Matériel Command

it was found that great quantities had been built up for an all-out assault at a later date.

B. The Enzian (Fig. 2) was a ground-to-air missile. This small-winged flak rocket, named after the flower gentian, was controlled in flight by radio, and was intended for use against heavy-bomber formations. Its plywood fuselage was 17 ft in length and carried 990 lb of explosive. Four auxiliary starting rockets were mounted outside the fuselage, and after 5 sec of flight were jettisoned. Plans were being made to modify the Enzian so that it could attain supersonic speeds, but because of the advances of the U. S. Army only a few successful test firings were made.

C. The Wasserfall was a ground-to-air missile. A 26-ft flak rocket, the Wasserfall, meaning waterfall, is very similar in design to the V-2 rocket launched against Britain. At first it was visibly controlled by radio from the ground, but later methods employed radar-tracking to guide it to the objective. Due to persistent allied bombing only 30 test launchings were made.

D. The X-4 was an air-to-air missile. It was a small rocket bomb, named Ruhrstahl or steel of the Ruhr, just under 6 ft in length, and designed to be launched from fighter aircraft. The rocket was controlled from the parent aircraft through two 4-mile lengths of wire. This fine wire was paid out from tapered bobbins enclosed in fairings at the extremities of two of the wings. Detonation occurred by means of a proximity fuse.

E. The Fritz X (Fig. 3) was an air-to-ground missile. This German bomb was released from aircraft flying at a minimum altitude of 22,000 ft. It was gyrostabilized and visibly guided into the target by radio. The Fritz X was ready for operational use in January, 1943, and was first employed successfully against our shipping and assault forces at Salerno. The warhead on this bomb was armor-piercing and carried a charge of 2530 lb of standard explosive.

F. The Hs-298 (Fig. 4) was an air-to-air missile. It was about 4 ft in length and could be controlled either by radio or wire. It was originally intended for use against sea targets, but early in 1945 it was being produced for air-to-air fighting in a factory in the underground railway in Berlin. A total of only 300 was built because the X-4 or Ruhrstahl bomb had become available.

G. The Hs-117 (Fig. 5) was a ground-to-air missile. It was named Schmetterling or butterfly, and was a rocket-propelled, radio-controlled missile to be launched from the ground-against-bomber formations. It accelerated to a speed of 560 mph, and was steadied in flight by a pendulum device. The take-off rockets burned out and were jettisoned; the main propulsion unit then drove the missile until it was detonated by a proximity fuse. Large-scale production began in January, 1945, in an underground factory at Nordhausen.

H. The Rheintochter (Fig. 6) was a ground-to-air missile. It was a rocket-propelled anti-aircraft weapon and was controlled in flight by radio. It traveled at a speed of 1100 mph and carried an explosive charge of 330 lb, equipped with a proximity fuse, to a ceiling of 48,000 ft. The starting rocket, attached to the base, was blown off after combustion was completed. Development did not go beyond the test firing stage, but experiments were still being conducted as late as February, 1945. The code name Rheintochter means daughter of the Rhine.

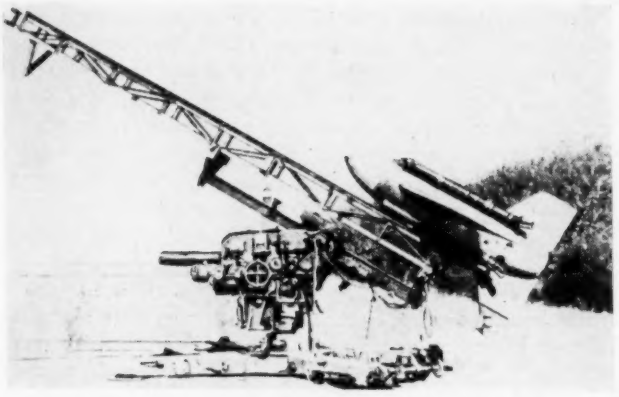
I. The FZG-76 (Fig. 7) was a ground-to-ground missile. It was launched either from the ground or from an aircraft. Ground launching as practiced by the Germans utilized a long run in which the working substance was the steam resulting from the catalytic action of calcium permanganate on a concentrated solution of hydrogen peroxide. The length of the gun and ramp varied from 140 to 170 ft at different sites. It carried 1870 lb of explosive, the range was 120 to 160 miles, and the circular probable error about five miles at a range of 130 miles when ground launched. The error was five times as great for air launching. The overall weight was about 5000 lb. The missile flew at an approximately constant altitude of about 2000 ft at speeds varying for individual missiles from 288 to 425 mph.

Control was by means of an autopilot monitored by a compass, and an altimeter. Fuel supply was controlled by means of an altitude and speed sensitive valve. The range was determined by an air log which shut off the fuel and set a dive-flap to dive the missile at a steep angle to the ground.

One of the firms interested in the development of these "Buck Rogers" devices experimented with nitric acid and mixtures of xylidines and amine compounds for fuel bases, developed powerplants for the controlled missile X-4; and also developed jet-assisted take-off units, commonly known as "jato" for the Me-262 and the Hs-117. This firm also worked on a motor for the Me-163 but the unit wasn't accepted because of delays in production and difficulties with the turbine drive.

Another firm interested in solid-propellant rocket motors for jato devices developed a monopropellant called "myrol" which consists of methyl nitrate. Numerous experimental powerplants were developed using gaseous oxygen and methanol or gasoline as the fuel components for use with

■ Fig. 1 – Beethoven was a carrier-controlled, crewless aircraft for one-time operational use against important, heavily armored and superarmored naval targets – composite aircraft consisted of Me-109 or FW-190 fighter mounted on top of Ju-88 bomber – flight performance corresponded generally to that of a bomber without load



■ Fig. 2 – Enzian was a small-winged flak rocket controlled in flight by radio

the Hs-293, a form of glide bomb with short-duration propulsion.

A third firm, designers of the Rheintochter controlled-missile series, and a fourth firm, designers of jato units, developed powdered coal inserts for a coal ram jet engine.

The Hermann Goering Aeronautical Research establishment located in Brunswick, set up a broad engine and fuel research program under a Dr. Lutz. Under his guidance the fuel system of nitric acid and visol (vinylethylether) was developed, together with rocket engines using the oxygen carrier nitrous oxide. This institution also worked on ram jet engines, and was instrumental in getting the firm of Bruessing to carry out experiments on the injection of nitrous oxide into the fuel of the Schmidt-Argus reed engine, the propulsion unit of the V-1.

The majority of the companies working in the rocket field were assisted by various engineering schools, for it was common for the developers to subcontract to small firms the detailed test work and the manufacture of small parts.

Numerous techniques for controlling the missiles were evolved, only one or two of which proved satisfactory. Conventional radar units, with special coding mechanisms were used. In one instance a beam-riding system was attempted, but the disastrous test results caused an immediate cessation of the project. It appears that the missile was supposed to follow a beam of energy emanating from a radar unit and aimed at the target, in this case an airplane; however, the missile would usually reverse its direction and ride the beam down to the radar unit, with quite obvious results. As a result, it was decided to augment the control system by adding homing or seeking devices of various sorts. These included systems sensitive to heat or sound. Some of the homing devices incorporated a miniature radar unit.

The German developments being discussed here are usually grouped under the loose term "guided missiles"; however, that isn't correct, inasmuch as some of the missiles are not guided and some of the developments are not missiles. I have already discussed another widely employed classification based upon the location of the launching device and the target; for example: air-launched-to-ground targets. The classification I shall employ is based upon the type of propulsion unit.

There are six general types of direct-reaction engines.

1. The first example is the reciprocating engine with the exhaust jet.
2. The second example is the gas turbine.
3. The third example is the turbo jet.
4. The fourth example is the reed, intermittent-combustion engine. This is better known as the "buzz bomb" engine. The reed engine is similar to the ram jet engine, with the exception that it is equipped with a grill at the forward end.
5. The fifth example is the ram jet, also known as the athodyd or the Lorin engine, after the inventor. This device consists primarily of a curved, barrel-like tube into which fuel and air are introduced.
6. The sixth and last example is the pure rocket. This is subdivided into the dry fuel and the liquid fuel subtypes. In either case, we have simply a combustion chamber in which the fuel burns. In the case of the dry fuel rockets, the fuel is also stored there. A great many of the German guided missiles were equipped with this type of propulsion equipment.

This discussion concerns itself with the latter three listed propulsion systems. For the purpose of simplicity, we will now consider the various engines and their applications, commencing with the reed engine and the ram jet. We have but one example of each of these, whereas we have some 30 examples of the rocket type.

The Schmidt-Argus reed or aeropulse engine was originally investigated by Messerschmitt and then given to Argus, which completed the development. This engine was something new when it appeared in London skies for the first time during the night of June 13, 1944. One Englishwoman, describing her experiences at seeing so many of them, thought they consisted of a glider train for aerial invasion. The blinking light in the rear she thought was a tail light.

The engine (Fig. 7) consists of a tube with an ingenious grill of bent strips of metal at the front end, which forms a spring-loaded valve. Following an explosion of the fuel-air mixture within the tube, the reduced pressure combined with the ram pressure against the front of the grill forces open the valve, permitting the entrance of air. Simultaneously fuel is injected. The following explosion closes the valve, discharges the products of combustion through the rear nozzle, and imparts a forward impulse to the weapon. The explosion frequency is on the order of 45 to 50 cps.

The life of the engine is 30 min, which is sufficient for the duration of the flight.

Subsequent to V-E Day, Prof. Schmidt demonstrated to Allied technicians that this engine could be constructed to develop twice the rated thrust without increasing the fuel consumption. This presents some idea of the degree of improvement of which this type of equipment is capable.

At least 175 of these weapons were constructed with cockpits and conventional controls. The pilot was to bail out just before striking the target, but it is believed that the piloted missile was not used operationally. The Japanese also had a version of the piloted V-1, Baka being the popular and news story name generally used. (See Fig. 8.)

Although the V-1 was relatively simple, let me emphasize the importance it achieved during the war. Between June 13 and September 3, 1944, approximately 8205 missiles were launched against England. Of these, 2354 landed in the greater London area, killing 5476 persons, destroying 23,000 houses, and damaging over 1,000,000 more. In addition to that, 1482 allied flyers were lost in bombarding the launching sites. The buzz bombs, then, were responsible for a number of deaths equal to approximately one-tenth of the number of Britons who lost their lives by conventional aerial bombardment; and this in the short space of three months and at very little cost in lives to the Germans. This points up emphatically the nature of the next aerial war. It is probably not an exaggeration to state that to achieve the survival of this nation we must in time thoroughly and effectively outlaw war, or remove our entire establishment from the surface of the earth. This is an alarming picture indeed, but not so alarming as the prospect of our total annihilation.

The next type of engine we will consider is the ram jet.

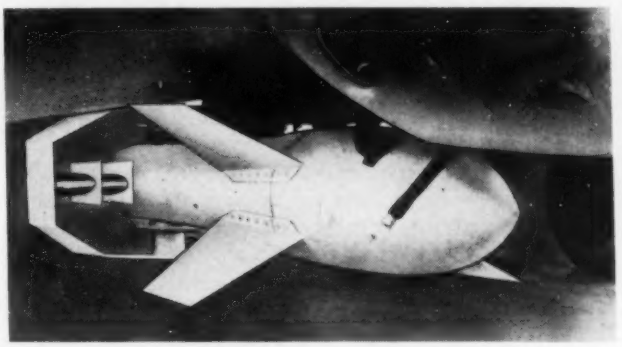
Only quite recently, now that air-speed values are approaching the velocity of sound, has it been practicable to envisage the use of the Lorin nozzle for propulsion. In the fall of 1944, while development in this country was limited largely to discussion as to whether such a power-plant would operate at all, the Germans had developed, built, and flight-tested a unit designed to produce a gross thrust of 4400 lb at 500 mph. (See Fig. 9.) This development was accomplished by Walter of Kiel, Dr. Saenger of the DFS (Institute for Study of Soaring Flight), and by engineer Pabst of the Focke-Wulf plant at Bad Eilsen. The Saenger design is characterized by a sharp inlet with a diffusing passage having a 12-deg included angle. The combustion chamber has a diameter of approximately 58 in. and the tail cone a divergence of 14-deg included angle. The total length is given as 23 ft. Approximately 60 fuel spray nozzles are located at the inlet end of the combustion chamber, and the spray is directed upstream to secure better mixing. For flight testing the athodyd was mounted above the fuselage of a Do-217 twin-engine plane and attached to the fuselage by means of thrustmeters. As the speed of this plane was insufficient to secure a good reading, it was dived to a speed of approximately 400 mph. At this speed a gross thrust of 3500 lb was realized. At speeds within the range of 250 to 500 mph the drag was found to be approximately equal to half the thrust. At full output the fuel consumption was on the order of 5-5½ lb per lb of gross thrust. At speeds of 500 mph the fuel consumption was equal to that of the Schmidt-Argus reed engine; at higher speeds it was better. At half the rated thrust the fuel consumption was 20 to 40% better than that of the Schmidt-Argus reed engine. Experience in this development indicated that it is not practical to build units with a gross thrust of less than 1000 lb, inasmuch as the relationship of surface to cross-section produces excessive drag.

A particularly effective employment of the Lorin tube is that of a jet propeller with a rotating wing. (See Fig. 10.) Here characteristics and performance values are very high; particularly in the range obtained at high altitudes. This is due to the fact that in rotating-wing aircraft the rotors can always be run at suitably high speeds. A rocket attachment can be provided to enable safe landing if the propellers are cut off. Take-offs and landings in these aircraft occur vertically, and only a small flying strip is required.

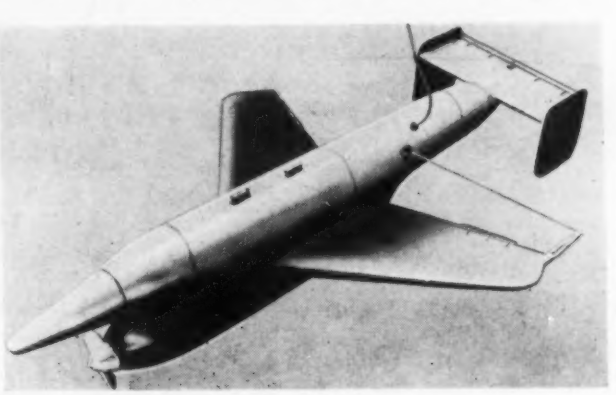
We will now consider in turn the various rocket engines and their applications.

The Enzian remote-control flak rocket (Fig. 2) was a rather interesting device. It was typical of many of their guided missiles inasmuch as it was radio-controlled, powered with a liquid-fuel rocket engine, and launched with the aid of four solid fuel jato units. An idea of the size can be gained by looking at the picture of the mount from which it was launched. The power units operated for a period of 62 sec, but their power was sufficient to launch the weapon at the speed of 1137 fps.

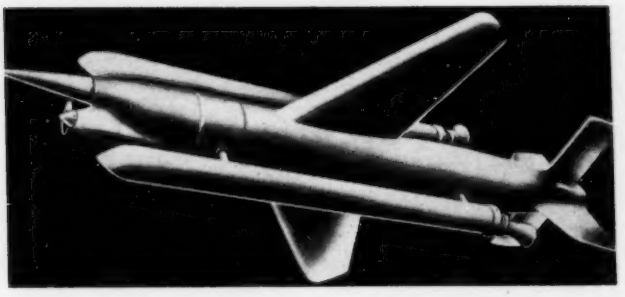
The two types of engines of this weapon were also typical of those of several of the other guided missiles. These engines were the Conrad and the Walter systems. The Conrad system had a pressure-fed fuel system; whereas the Walter unit had a pump-fed fuel system. The Conrad unit had the advantage of producing maximum thrust in



■ Fig. 3 – Fritz X mounted on carrier aircraft – remotely controlled (by wire or radio) bomb for use against armored naval and land targets – carried 2530-lb charge, which was capable of piercing 28-in. armor plate from bombing altitude of 19,700 ft



■ Fig. 4 – Hs-298 remotely controlled (by radio or wire) fighter weapon – explosive weight: 55 lb – attacking distance: 0.3 to 1.5 miles



■ Fig. 5 – Hs-117 (Schmetterling) radio-controlled flak rocket used as ground defense against enemy aircraft – liquid drive unit with solid fuel starting aid – explosive weight: approximately 50 lb

one second, but the disadvantage of being somewhat heavier than the Walter unit, which, although lighter was more complicated, thus increasing the chance of failure. A pump unit requires 3-5 sec to produce maximum thrust, since the turbine must have time to develop speed. For this reason we usually find pump systems in that type of missile which consumes a great deal of fuel and operates for a longer period of time. The classic example of this is the A-4, or, as it is more commonly known, the V-2.

The typical pressure system is equipped with an air-bottle pressurized, in the case of the Enzian, to 200 atmospheres. The pressurized air forces fuel and oxidizer from their separate tanks into the combustion chamber through an injector, where the fuel and oxidizer impinge and ignite spontaneously. In the case of the Conrad powerplant the oxidizer flow was routed through the combustion-chamber cooling jacket for proper cooling of the combustion chamber.

Incidentally, this particular operation of cooling the combustion chamber is one which the German scientists never were able to solve satisfactorily.

The proper functioning of the various types of rocket engines is dependent upon the temperature of the combustion chamber. This is influenced by the speed of the missile, the density and humidity of the atmosphere, the speed of fuel flow, and the length of the combustion chamber (this varies in the case of the solid fuel unit), to mention the most important considerations. These factors alter considerably during the flight of each of the missiles.

Several ingenious methods were employed to offset as much as possible this disadvantage of variation in the combustion-chamber temperature.

The combustion chamber of the X-7, for example, was divided into what amounted to two chambers. This weapon burned solid fuel, and the second chamber had a discharge nozzle which ran through the first chamber. Thus, the second chamber discharged its gases without their being cooled by the walls of the first chamber.

The A-4 employed a more complicated system. The coolant jacket surrounding the combustion chamber had some dummy passages through which the coolant did not flow until the temperature was sufficient to melt certain metal plugs which acted as stoppers.

Before I digress too much, however, let me return to the typical powerplants which I was discussing. I have already described the typical pressure-fed fuel system.

The pump-fed fuel system begins with an air-bottle which has air under pressure. When the engine starts this air forces T-Stoff (hydrogen peroxide) into the catalyst chamber wherein steam is generated. This steam is forced into a turbine which in turn operates two pumps, each pumping one of the two propellants. The propellants flow into the combustion chamber and there ignite.

It is as simple as that in theory, but in practice there were many complications. The acids had a tendency to corrode the tanks and pipes. This was overcome by not filling the tanks until just prior to launching.

Any sort of leak promised immediate disaster, since the propellants were explosive when united. The great pressure in the lines and the terrific speeds of the moving parts made them subject to failure. In operational use a failure could be a serious thing. Explosions on the ground were frequent. The dismantling of a rocket engine that has malfunctioned is a job no one wants.

The most highly publicized missile of the Germans was the V-2, or as the Germans themselves knew it, the A-4.

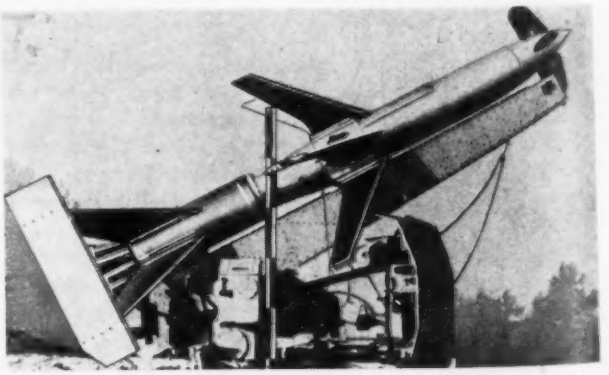
There were 10 variations of this weapon. (See Fig. 11.) Three were prototypes. Four were improved models. One was a launching device intended to increase the already great range to 5000 km. Interestingly enough, one of the improved models was to be equipped with wings, wheels, a pressurized cabin, and to carry a pilot. Contrary to what may be popular opinion, the launching acceleration was well within the limit which a human can survive. A-4 moved the first 50 ft at an average speed which a good track man can maintain without difficulty.

The "A" series was considered to be an intermediate step to the practical use of guided missiles. The inherent shortcomings of this type of weapon were partly conditioned by the war, but the future will disclose improvement occurring more rapidly than was thought possible prior to the war.

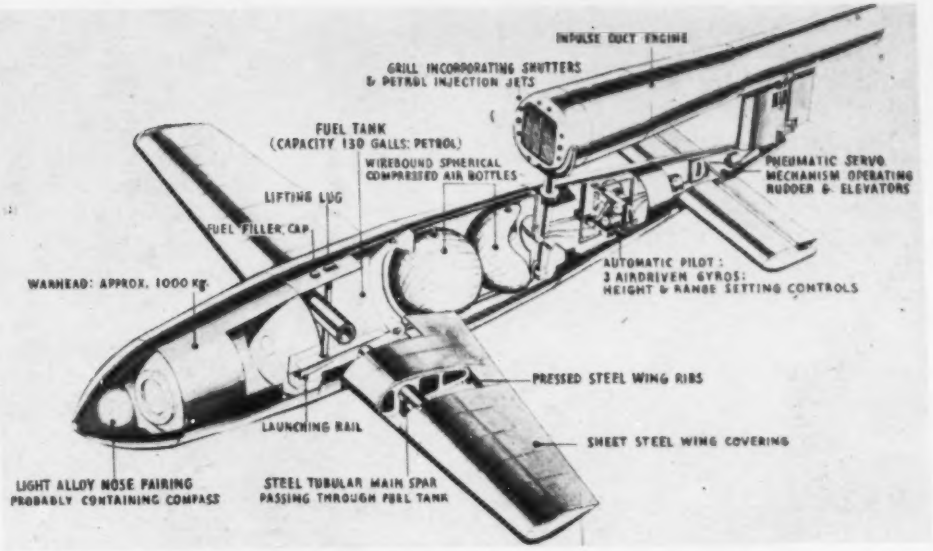
The 13.5 tons of this missile were pushed through the air for a distance of 230 miles at a maximum speed of 3735 mph. That is approximately three times the speed of rotation of the earth at the equator.

This tremendous speed was made possible by the ravencus liquid fuel pump-fed rocket engine, which consumed a total of 8800 lb of alcohol and 1093 lb of liquid oxygen in the short span of 65 sec, and which developed a maximum thrust of 68,500 lb.

It is interesting to note that this missile was extremely expensive, each requiring 20,000 man-hr for construction; however, since the Germans launched 3165 A-4's operationally, it is clearly indicated that the place of the large rocket weapon is firmly established in modern warfare. It has been pointed out that the small size of the warhead – 1 ton – would hardly seem to have made the cost of the weapon worth while. In this connection it must be pointed



■ Fig. 6 – Rheintochter I was radio-controlled flak rocket used as ground defense against enemy aircraft


IMPULSE DUCT ENGINE
GRILL INCORPORATING SHUTTERS & PETROL INJECTION JETS
FUEL TANK (CAPACITY 130 GALLS: PETROL)
WIREBOUND SPHERICAL COMPRESSED AIR BOTTLES
PNEUMATIC SERVO MECHANISM OPERATING RUDDER & ELEVATORS
LIFTING LUG
FUEL FILLER CAP
WARHEAD: APPROX. 1000 Kg.
AUTOMATIC PILOT: 3 AIRDRIVEN GYROS: HEIGHT & RANGE SETTING CONTROLS
PRESSED STEEL WING RIBS
LAUNCHING RAIL
SHEET STEEL WING COVERING
LIGHT ALLOY NOSE FAIRING PROBABLY CONTAINING COMPASS
STEEL TUBULAR MAIN SPAR PASSING THROUGH FUEL TANK

■ Fig. 7 – FZG-76 was the compass-controlled long-range bomb known as V-1 – catapult launched from stationary platform – operational use against surface targets – weapon was 27.3 ft long and had a wing span of 17.6 ft

out that it was probably the intention of the Germans that the weapon would eventually carry some sort of atomic device, in which case the warhead would have had the requisite specifications.

Another example of the A-4 type of missile was the Wasserfall. This was a supersonic, radio-controlled flak rocket bearing a great resemblance to the A-4. The three principal points of difference were: the addition of four stubby wings to the Wasserfall, the substitution in the Wasserfall of a pressure-fed fuel system, and size: the Wasserfall being approximately one-half the size of the A-4.

This weapon gave great promise of success, and at the end of the war it had the highest priority for development.

Another pair of missiles employing the Conrad pressure-fed power plant was the Rheintochter I (Fig. 6) and its development the Rheintochter III. Both of these missiles were similar in all respects to the Enzian except in the wings and control surfaces, and in the jato units.

One of the German devices was the composite aircraft Beethoven, consisting of an Me-109 or an FW-190 attached to a Ju-88. (See Fig. 1.) The pilot of the fighter plane controlled both aircraft directly during take-off and the run to the target area. Upon reaching the target area, the explosive-loaded Ju-88 was released and controlled by radio from the fighter plane. This device was not particularly satisfactory because of the slow speed and vulnerability of the Ju-88.

The guided missile X-4 was an interesting device. This was a liquid fuel rocket-powered subsonic wire-controlled air-to-air weapon. It was approximately 6 ft in length. The powerplant employed the pressure-fed system. Two of the four stabilizing fins were equipped with wire spindles. When the weapon was launched from the parent aircraft the wire was unreeled. Control was effected from the parent aircraft electrically through these wires.

The 20-kg blast-effect warhead was equipped with a combined acoustic-proximity fuse to detonate at 50 ft.

Another most interesting device was the Hs-117. (See Fig. 5.) A variation of this was the Hs-117H. The Hs-117 was ground-launched with the aid of two jato units. The Hs-117H was air-launched from an aircraft. Both units were guided missiles in the true sense.

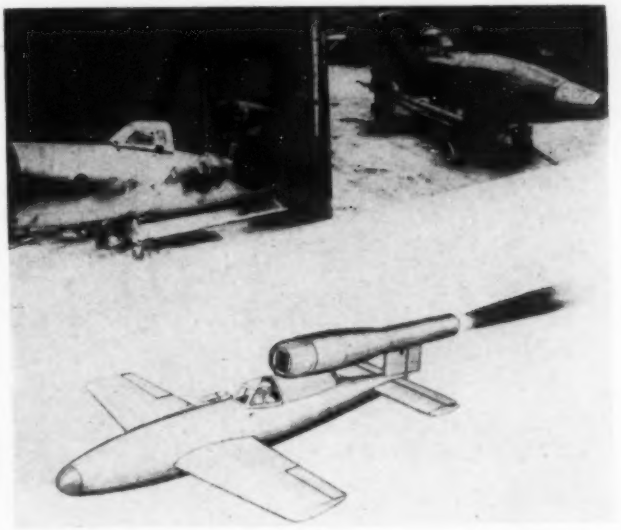
The weapon itself was 14 ft in length. The peculiar nonsymmetrical nose was so designed as to permit the installation of the fuse on one projection and the propeller-driven generator on the other. The generator supplied the power for the radio control system.

The jato system on the ground-launched version is interesting. The lower-mounted unit was ignited first. The thrust developed threw the missile free of the launching cradle. When this was accomplished the upper-mounted jato unit ignited. When both jato units were expended they were jettisoned with the aid of an explosive charge.

The power system of the missile proper employed a typical pressure-fed rocket unit. It developed a thrust of 370 kg for 50 sec. This was sufficient to drive the missile's 968 lb at a speed of 470 mph to a maximum altitude of 35,000 ft.

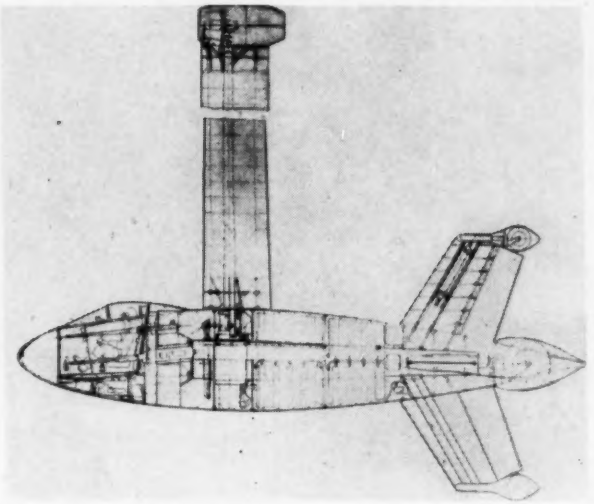
Another weapon, the Hs-298 (Fig. 4) was a scaled-down version of the Hs-117H and was intended for use as a plane-launched air-to-air missile. This weapon was 4 ft in length and weighed 264 lb. It used a solid fuel jato power system. This weapon was inferior to our own aircraft-launched rockets.

The introduction into the war of the Hs-293, Hs-294,



■ Fig. 8 – Japanese piloted version of German V-1 bomb

■ Fig. 9 – Performance trials of Saenger-Lorin tube mounted to fuselage of Do-217



■ Fig. 10 – Sketch of proposed rotary-wing fighter aircraft equipped with Lorin propellers (one of three wings shown)

Hs-295 series brought about the first allied casualties caused by secret weapons and started a wave of speculation concerning the nature of future aerial war which will probably never subside. The imaginative articles which were written about these weapons, however, can never equal the thrilling report which came from a British merchant convoy in the Mediterranean Sea in October, 1943. A number of these odd-looking and strange devices were launched against men totally unprepared for such a surprise. The calm courage they displayed in warding them off with the aid of conventional AA guns, not knowing what unusual behavior the weapons might exhibit next, was a tribute to the courage of the men.

Between the time of this initial attack and February, 1944, 159 of these missiles were dropped in the course of 12 attacks in the Mediterranean, resulting in five vessels sunk and two damaged.

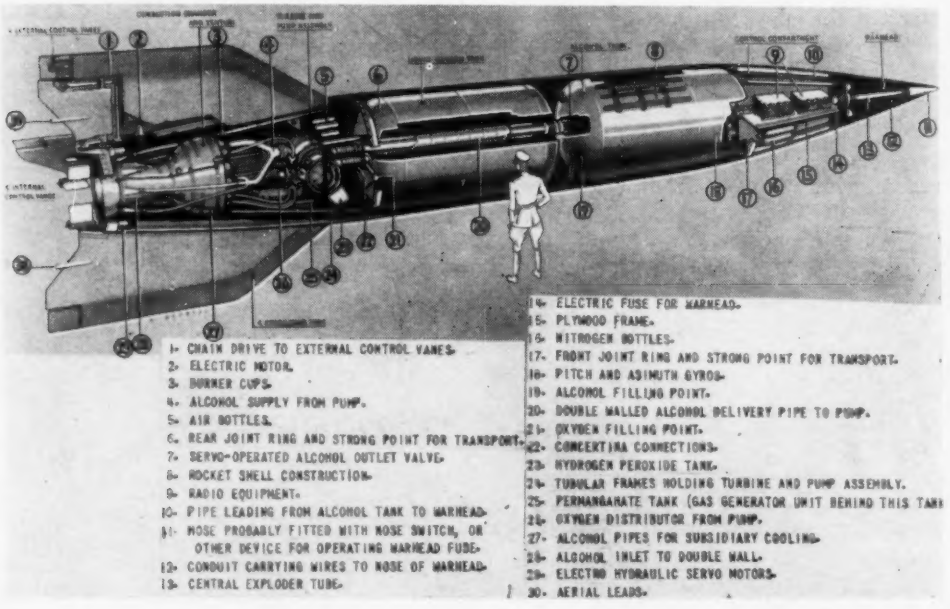
Inasmuch as air superiority is necessary for the use of guided missiles, not much was heard from them in the latter stages of the war, but the Germans' entire effort in this field was justified if only their success in the Mediterranean is considered.

The Hs-293, Hs-294, Hs-295, and their 47 modifications were designed to be propelled by liquid fuel rocket units. They were of the typical pressure-fed system. (See Fig. 12.) Since the missiles were launched and controlled by wire or radio from the launching aircraft, the purpose of the jet units was to drive the missiles ahead of the launching aircraft in order to permit the controlling operators to see and direct them toward the targets.

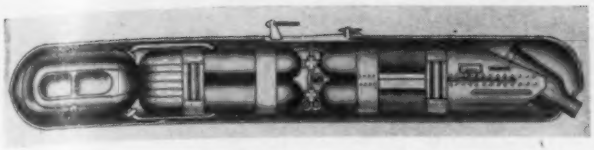
The Hs-293 (Fig. 13) was 13 ft in length and weighed 1 ton. The Hs-295 was similar in size and performance. The Hs-294 was a scaledup version of the Hs-293. This weapon was 20 ft in length and weighed somewhat over 2 tons. The maximum speed possible was 482 mph for a range of 9 miles. It was designed to carry a greater load and was powered with twin rocket engines. The airframe was designed to be sheared off upon contact with the sea. The torpedo was then to travel in the conventional manner.

Among the items being discussed only one was a bomb in the pure sense – that is, without auxiliary power system. That was the Fritz X, 1400-kg bomb. (See Fig. 3.) This weapon was remotely controlled, however, in azimuth and range by the addition of radio-controlled rake-spoilers located and operating continuously on the tail surfaces. The antennas were located along the leading edges of the stabilizer. One version of this bomb was wire controlled.

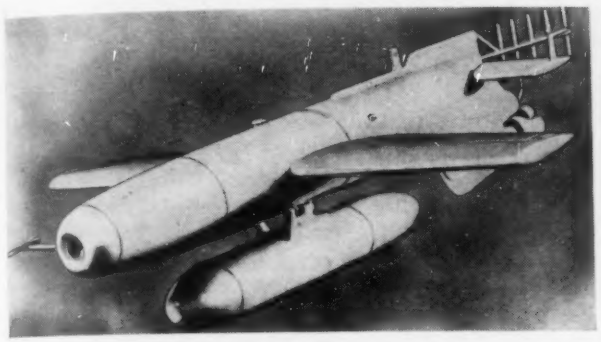
One of the Germans' most fascinating projects was their long-range bomber. (See Fig. 14.) This was a liquid fuel supersonic pilot-controlled aircraft intended to fly from Germany to New York in 40 min at an altitude of approximately 154 miles. The motor was to weigh 2½ tons and to deliver a thrust of 100 tons. This bomber was never


1- CHAIN DRIVE TO EXTERNAL CONTROL VANES.
2- ELECTRIC MOTOR.
3- BURNER CUPS.
4- ALCOHOL SUPPLY FROM PUMP.
5- AIR BOTTLES.
6- REAR JOINT RING AND STRONG POINT FOR TRANSPORT.
7- SERVO-OPERATED ALCOHOL OUTLET VALVE.
8- ROCKET SHELL CONSTRUCTION.
9- RADIO EQUIPMENT.
10- PIPE LEADING FROM ALCOHOL TANK TO WARHEAD.
11- NOSE PROBABLY FITTED WITH NOSE SWITCH, OR OTHER DEVICE FOR OPERATING WARHEAD FUSE.
12- CONDUIT CARRYING WIRES TO NOSE OF WARHEAD.
13- CENTRAL EXPLODER TUBE.
14- ELECTRIC FUSE FOR WARHEAD.
15- PLYWOOD FRAME.
16- NITROGEN BOTTLES.
17- FRONT JOINT RING AND STRONG POINT FOR TRANSPORT.
18- PITCH AND AZIMUTH GYROS.
19- ALCOHOL FILLING POINT.
20- DOUBLE WALLED ALCOHOL DELIVERY PIPE TO PUMP.
21- OXYGEN FILLING POINT.
22- CONCERTINA CONNECTIONS.
23- HYDROGEN PEROXIDE TANK.
24- TUBULAR FRAMES HOLDING TURBINE AND PUMP ASSEMBLY.
25- PERMANGANATE TANK (GAS GENERATOR UNIT BEHIND THIS TANK)
26- OXYGEN DISTRIBUTOR FROM PUMP.
27- ALCOHOL PIPES FOR SUBSIDIARY COOLING.
28- ALCOHOL INLET TO DOUBLE WALL.
29- ELECTRO HYDRAULIC SERVO MOTORS.
30- AERIAL LEADS.

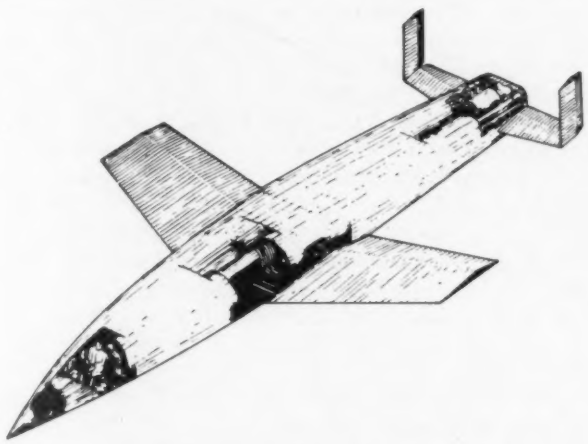
■ Fig. 11 – A-4 rocket (known as V-2) was a large, fin - stabilized, rocket-propelled missile, one type of which was radio controlled, the other type containing no radio, but being capable of complete self-guidance by use of gyroscopic integrating accelerometer – missile was 46 ft long and 5 ft in diameter – warhead weighed 3150 lb. It attained a speed of 5230 fps at end of burning, which fell to about 2500 fps in its descent through earth's atmosphere – since its speed was supersonic, nothing was heard until it had landed and exploded



■ Fig. 12 – Pressure-fed system power unit for Hs-293H glider bomb – thrust of 1340 lb delivered for 11 sec – no fuel pump is required for this apparatus, fuel flow being accomplished by gaseous oxygen – operation was guaranteed at temperatures ranging from 158 F down to —76 F without heating



■ Fig. 13 – Hs-293D radio-controlled glider bomb (with television installation) was launched from carrier aircraft for use against land and nonarmored naval targets – maximum speed: 536 mph – television picture was not effective until bomb was 2.5 miles from target – television installation was still in developmental stage



■ Fig. 14 – Rocket bomber designed to fly from Berlin to New York in 40 min at an altitude of 154 miles – engine burned liquid oxygen and alcohol – rocket nozzle was cooled by water, condenser being used to form water from products of combustion – other side of condenser could be used to vaporize the alcohol. Test model was made that carried one man and had landing gear, although it is not known if this model ever flew; it is known, however, that test runs were made on its engine

finished, but it is believed that time was the only obstacle against its completion.

This bomber was to be catapult-launched at 500 mph and rise to altitude in 4-8 min, during which time the fuel would be exhausted. It was then to glide and skip along the outer atmosphere with decreasing oscillations. The Germans hoped to be able to destroy any large city on the earth with a fleet of 100 of these bombers within the space of a few days' operations.

In conclusion may I state that the Germans in the guided missile field were 10 years in advance of similar American development.

It will be noted that the Germans very largely explored the entire field of rocket-powered equipment. Their program was just beginning to become stabilized with the production of those units which were adaptable to mass production and operational use.

Very fortunately, they were never able to obtain full use of their developments. We are all familiar with the immense rocket-launching establishment on the coast of France which was to be used in destroying London.

It is now the responsibility of the American industrial machine to begin where the Germans stopped and to provide ourselves with the equipment necessary to maintaining our leadership in the scientific world.

American industry and the War Department must cooperate in exploiting this field, in which there appear to be unlimited possibilities for new and better devices and equipment, not alone in the province of weapons of war, but also in the applications to a fuller and more satisfactory civilization.

Those actively interested in investigating any of the developments I have mentioned, as well as any other development of any of the warring powers, and who wish to obtain technical information on these developments may do so by applying to the Air Matériel Command at Wright Field. There, under the direction of General Nathan F. Twining, Commanding General of the Air Matériel Command, there is in operation a technical organization which is charged with the responsibility of procuring anything you may require along this line.

Please feel free to call upon us at any time for whatever foreign information is available that will assist you.

The Command is supported by the American taxpayer not to procure information and place it under lock and key but to assist you to the fullest extent in our common effort of preserving the peace of the world, maintaining our high standards of civilization and culture, and in maintaining our military supremacy.

DAMPING IN SUSPENSIONS

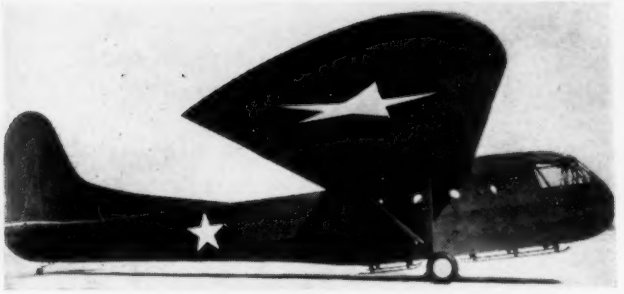
continued from page 393

sembling. Friction existing in new installations may vary with wear, lubrication, or changes in temperature. We have all had experiences in which otherwise good rides were thrown off by one or more of these variables.

The specification for shock absorber damping must then be predicated on conditions selected as average and with consideration at least for effective damping of movements of the unsprung weight and of the car body. These conditions being different in various constructions requires the specification of different types of control. The author refers to the inertia-control shock absorber as an approach to the ideal for control of body motion. But if adequate control of the unsprung mass is provided the inertia device must be supplemented by additional control – this, in some cases at least, being supplied by special valves not under the control of the inertia element.

That we may not have given as much attention as we should to the type of damping incorporated in the shock absorber becomes more and more apparent as we study Mr. O'Connor's paper. To a considerable extent we can agree with the author that current types of shock absorbers operate on the same principle with valvings to introduce different resistance characteristics. The need for greater comfort, better handling and safety, and reduction of car vibrations of all sorts makes it imperative that we give the factors discussed by the author a more thorough examination to see if there are not greater possibilities in our present units and what new constructions would be necessary to do a better job.

ADHESIVES—



■ Fig. 1 – Invasion glider that contains numerous parts fabricated with adhesives

ADHESIVES have served the war effort well. The record of duty performed is enviable and practically without blemish, and in many cases surprising, because on a number of applications adhesives did not enter the picture until other fabricating methods had failed. Development of some of the component parts of aircraft and radar equipment would have been very seriously curtailed without the special characteristics offered by cements and cementing techniques. It is inevitable that these characteristics and others be contemplated for civilian fabrication. (See Figs. 1-3.)

The necessarily large volume of data collected during the last five years is rapidly being compiled and tabulated, and soon will be available to designers. These data will contain, besides the usual shear, tension, fatigue, and stress-strain information on laboratory specimens, a considerable quantity of field data as well as dynamic and static stress data on composite structures.

THE application of high strength adhesives to the fabrication of automotive parts and assemblies is discussed here by Mr. Swayze, who covers the design, tooling, and inspection phases of the subject, the properties of adhesives, and the relative cost of chemical bonding, compared with other methods of assembly.

To benefit fully from the use of adhesives, the author reports that the parts must be designed so that loads applied to the bond are in shear or direct tension, eliminating peel and cleavage.

Freedom of design, he added, however, is permitted by the wide variety of materials that can be bonded in all sorts of combinations.

THE AUTHOR: D. L. SWAYZE, chief chemist of the Cycleweld Division of the Chrysler Corp., has been connected with the research department of that organization since 1937. His early work consisted of the development of Cycleweld Cement and allied adhesives and cementing processes. A graduate of Michigan State College in 1935, he was an instructor there in bacteriology in 1936.

[This paper was presented before the Detroit Section of the SAE, Detroit, Nov. 1, 1945.]

To the production man will be offered the benefits of considerable production experience on a relatively new procedure of structural attachment, and to the sales group will be offered numerous selling points, a few of which are light weight, greater strength, and general attractiveness.

In anticipation of the use of adhesives to replace conventional methods for any assembly, the designer would first weigh the advantages gained against the disadvantages. The advantages usually claimed for bonded structures are greater joint strength, ability of the bond, with its elastic characteristics, to absorb impact and dynamic fatigue stresses, the ability of the bond continuity characteristics to increase stiffness in bending and to allow maximum efficiency of bond per unit bond area. Shorter, more efficient production schedules are possible in some cases, a wide variety of materials that can be bonded broadens design possibilities, and decrease in structural weight is realized. It can probably be said that the greatest advantage of adhesive attachment is the benefit gained from the continuity of the bond. Here the distribution of load, or the elimination of concentration of stress, offers a long-sought solution to crippling or ultimate buckling stresses, prevalent when attachment is at only intermittent points. This may be translated in a simple structure, such as a stiffened sheet panel, to ability to carry higher compressive loads and ability of the bond to prevent initial torsional instability failures of the stiffener. The above characteristics are realized in actual tests. The stiffness of the continuous bond also may be directly translated to lighter component parts, resulting in decreased basic structural weight. Perhaps the second most important and very closely allied property of the adhesive bond is the low modulus of elasticity of the cement. Absorption of impact, fatigue, and shear stresses in the bond allow broader design possibilities than are encountered with conventional methods of attachment.

The disadvantages are mainly in tooling, inspection of bond, and field repair of damaged structure. The problems of tooling and inspection of the finished bond will be handled later. Maintenance and repair of field damage may offer some difficulty to the practical minded designer. Of course, the replaced parts may be riveted or bolted into place, however, in several instances repair fixtures and kits have been used to reestablish cemented bonds in the field. In general, it has been found that a great deal more difficulty has been encountered in deliberately attempting to fail bonds than has been encountered in attaining the bond initially.

With respect to materials capable of being bonded, practically none of the structural materials encountered in present-day fabrication has been found incapable of being bonded. Wood, metals, thermoplastics, and thermosetting plastics, ceramics, fibers, rubber, glass, and many other

MODERN TOOL OF FABRICATION

by D. L. SWAYZE
Chrysler Corp.

materials have been bonded to themselves and in combination through the use of adhesives of one type or another. (See Figs. 4-6.)

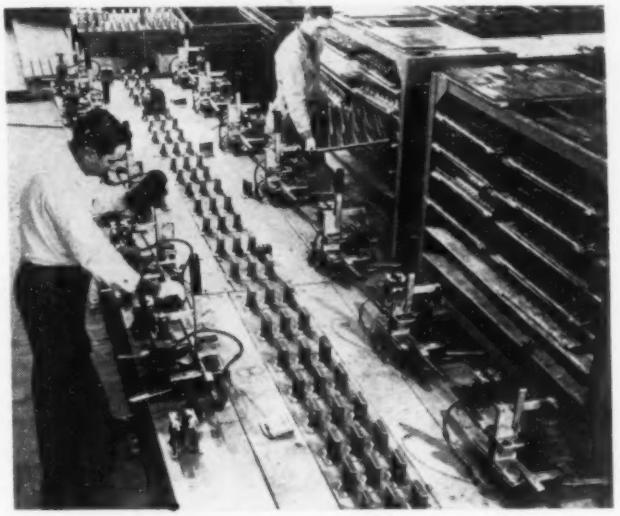
Probably the second consideration of the designer would be that of physical properties of the adhesive to be used. These data consist of tensile shear, compression shear, tensile, shear impact, tension impact, tension impact fatigue, and fatigue in vibration on several types of specimens and at various temperature conditions. Additional data are available on cleavage, peel, and flexual strength, and resistance of bonds to chemicals, solvents, oils, aging, oxygen, salt spray media, and climatic conditions at various testing stations throughout the continent. Standardization of tests and new testing methods are being developed by ASTM Committee D-14, and after the recent meeting at Atlantic City, it can be reported that excellent progress is being made by this very capable group of people towards writing standard adhesive tests.

The utility of any cement is, of course, influenced to a large extent by the strength of the bond it will give and the permanence of that bond in the conditions under which it is used. It is desirable that the shear and tensile strength of the bond be as great as the strength of the material bonded. In wood adhesives this is easily accomplished, but with metal-to-metal adhesives this can be realized with practical overlaps only in the thinner gages, because the tensile-shear strength of the cements is less than the tensile strength of the metal, running from 3000 to 4000 psi. The compressive shear or block shear strength is considerably higher than this, due, of course, to the method of testing, but this type of loading is encountered only occasionally in practice. The bond shear strengths compare very favorably with the strength of spot welds and rivets in thinner gage material on the basis of equal area of overlap, while heavier gages of metal may be riveted or bolted to give strengths higher than the 3000-4000 psi limit realized with adhesives. It should be mentioned at this point that tensile-shear strength is not directly proportional to depth of lap, a 1-in. lap having strength less than twice that of 0.5-in. lap. This is tied up with the fact that in loading a lap joint, a large proportion of the stress is taken up on the edges of the lap.

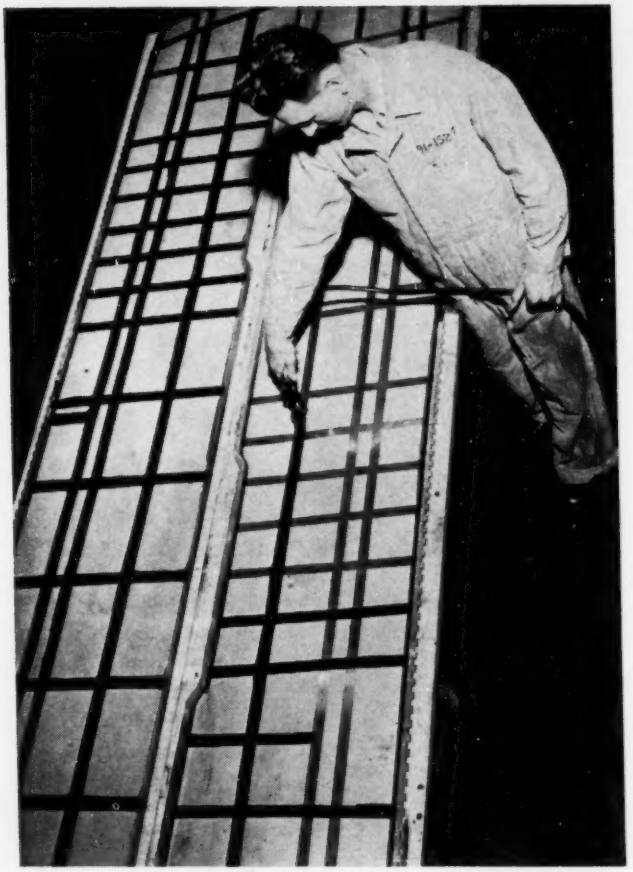
The tensile strength of wood-to-wood bonds is under investigation at the present time; five different types of commercial wood adhesives have shown tensile strengths of 600 to 1100 psi on maple. The metal-to-metal bonds have tensiles in the range of 3000 to 5000 psi, and in isolated cases, somewhat higher values have been obtained. Peel strength and cleavage are related to the tensile strength and, in considering high strength metal adhesives, it has been found that peel and cleavage strengths are low compared to shear. The influence of this on design will be covered a little later.

In most bonded structures a sudden application of load may occur occasionally, or the bonds may be subjected to alternating stresses, such as vibration. It is, therefore, necessary that impact strength and fatigue properties of the adhesives be determined. The impact picture has been confused by lack of a standard specimen. We have been developing and testing a block shear impact specimen for ASTM Committee D-14, which has given us values of 15 and 25 ft-lb per sq in. on two metal-to-metal adhesives with which we are most familiar, and 10 ft-lb per sq in. with a well known resorcin glue on birch and maple. A tension-impact test is also under observation which gives lower foot-pound values, and may also be adapted to tension-impact fatigue.

Performance under alternating stresses may be predicted from fatigue data, but this picture is also confused by lack of standardization of test method. One suitable method involves the application of a static load in shear, superimposed on this is a dynamic load, applied and released 3600 times per min. By means of this type of test, it has been demonstrated that at least some of the metal adhesives are better in fatigue and vibration than rivets and spot



■ Fig. 2 – Multiple bench fixtures used to fabricate ski pedestal for glider shown in Fig. 1 – finished article (next to fixture, lower foreground) consists of two blocks of synthetic rubber cemented to steel male and female sections so mount acts in shear to absorb landing shocks of glider



■ Fig. 3 – Method of application of cement utilizing flow gun – parts being coated are skin section of P-4ON wing flap

welds. This has been confirmed by tests on fabricated parts.

Another physical property of an adhesive bearing directly on the life of a bonded assembly is the amount of creep in the cement under sustained load. Obviously, if a material continues to creep under a sustained load of the magnitude encountered in a bonded assembly, it will eventually fail. This is determined by measuring deflection over a long-term static loading. All adhesives take an initial deflection on application of load; however, some level off at this point, in others the deflection continues to increase. The load deflection and hysteresis curves are also of value along this line.

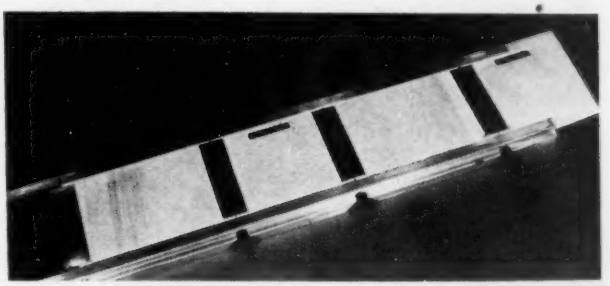
Most bonded assemblies are required to perform under varying temperature conditions. The strength tests that have been covered should be run at elevated and subzero temperatures as well as at room temperature. In general, the metal-to-metal adhesives lose some strength at 160 F; some increase, others decrease in strength at —70 F.

Permanence tests of importance include water and humidity resistance, cyclic tests involving heat and water, salt spray, exposure, long-term aging, resistance to gasoline, oil, and so on. Other tests may be necessary in particular applications.

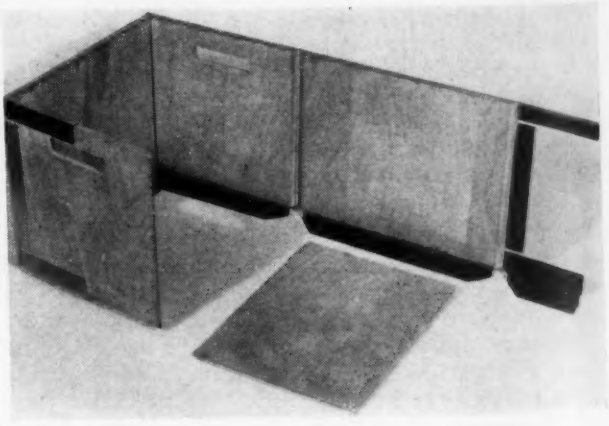
At the present time no one particular metal-to-metal adhesive excels over all others in strength and permanence properties, and in choosing an adhesive for a certain application, one must compromise on some features.

Aside from the performance of the finished assembly, which may be in part predicted from the test data we have discussed, one must also consider what is involved in application of the adhesive and preparation of the final bond. All adhesives require that the material to be bonded be clean. Various methods, including sand blast, vapor degrease, alkaline cleaning, and pickling on metal, and planing and sanding on other materials, have proved satisfactory and can be adapted to large-scale production. The application of the adhesive material, of course, depends on the viscosity, solid content, tack, and flow, and suitable methods include glue spreader, spray, flow guns, and dipping. After application of the adhesive an open time or drying period is usually required to eliminate solvent or to stiffen the adhesive, which depends on the volatility of the solvent, reactivity of the glue, thickness of film, penetration, and so on. This period with various adhesives varies from 5 min to 1 hr or longer. After drying, the parts are then assembled, placed in a suitable pressure fixture, and heat and pressure applied to cure the adhesive. The curing time can be shortened to increase production per fixture if the materials being bonded can tolerate elevated temperatures. For instance, if the cure time is 15 min at 325 F, the same cure might be obtained in 2 min at 400 F. The cleaning, coating, and curing varies with the adhesive used, materials bonded, and the design of the assembly, so we have attempted to give merely a bird's-eye view of the general process.

In regard to designing for adhesives, one cannot be merely told how or how not to design. I can, however, tell about some of the principles that we have learned about the problem. In the case of Cycle-Weld, most of the work done on development has been done during the war, when automobiles were out of production, thus, unfortunately, the learning and experience were not gained on



■ Fig. 4 – Hollow container prefabricated in the flat



■ Fig. 5 – Forming of hollow container

automobiles. This, however, was true with most war products, and while most of us were working on tanks, planes, and guns, we were thinking a little about automobiles, or at least how the things we were using or making could be applied to them. (See Figs. 7 and 8.)

It became clear after the first few assemblies that in most cases, in order to gain full advantage of the adhesive, one must design for it. In most cases adhesives have their own tooling requirements and strength limitations, and cannot be adapted most efficiently to assemblies designed for other methods of fabrication.

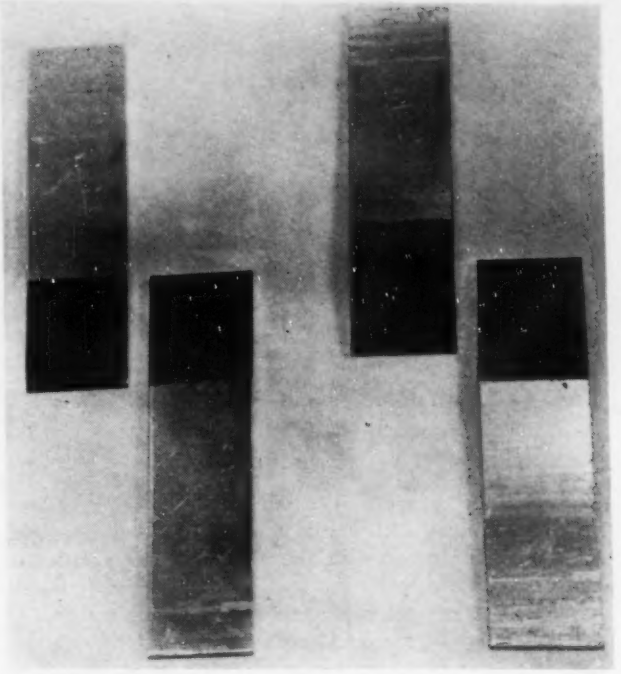
It has been found that assemblies should be designed so that the bond is in shear or direct tension, eliminating peel and cleavage. Any bending in a sheet that is concentrated at the edge of the bond is bad because of peel. In aircraft structures that were bonded, although not designed for adhesives, the judicious use of a rivet at the end of long seams has averted trouble from peeling.

The strength of a shear lap joint varies directly with the width of the bond, but, as stated before, strength is not directly proportional to depth of lap. Thus, the most economical way to increase strength is to widen the lap, rather than increase the depth.

In the case of metal-to-metal joints, it becomes necessary to design for relatively light gages. The reason for this is that heavy fittings usually mean high stresses. As loads become more concentrated, mass is emphasized and bondable area is reduced. In such cases the external load requirements often exceed 3000 psi, the present limit of a Cycle-Weld bond. In bonds between heavy sections where the surfaces to be bonded are not perfectly mated, residual stresses in the parts, due to deformation to obtain contact for bonding, may produce internal loads higher than 3000 psi. These internal stresses may be minimized through the use of adhesives that bond at low pressure. (See Fig. 9.)

An important factor in the design of parts held together with nuts, bolts, and rivets, is often the bearing strength of the material at the point of fastening. In a bonded joint, however, the continuity of the bond makes the whole bond area work. Gages, therefore, can often be reduced without any sacrifice in strength of assembly.

■ Fig. 6 – Finished hollow container



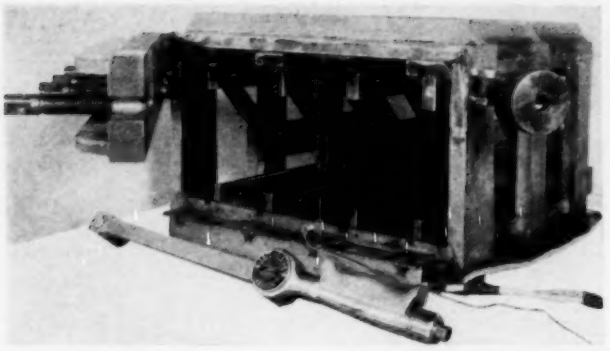
■ Fig. 7 – Typical alclad Cycle-Weld laboratory test specimens showing both good and poor contact

Freedom of design is permitted by the wide variety of materials that can be bonded in all sorts of combinations. Wood, metal, rubber, plastics, fibers, and ceramics can be used in new and better engineered designs with the help of adhesives.

Having discussed the characteristics of the bond, we will now look into applications of adhesives to automobiles. The first thing that comes to mind is cementing rubber motor mounts to metal, which can now be accomplished without brass plating the metal and bonding during vulcanization. The field of bonding to other materials than metal is also opened. The bonding of wood to metal certainly has a great many applications on the station wagons, and will improve that product considerably. Plastic tops, laminated with adhesives, can be used on station wagons and other models where low production limits die expenditures. There is also a field in attaching parts to the inside of the body where we must preserve a clean, smooth surface on the other side to minimize metal finishing. Applications of this type that have been worked on include cowl ventilators and deck lid stiffeners. These and many others are applications to the present automobile.

There will be many more in the car of tomorrow. For example, the continuity of the bond and the ability of adhesives to bond unlike materials make it possible to exploit the moment of inertia. This is one of the greatest contributions adhesives have made to engineering, namely, the stressed-skin principle as applied through the medium of sandwich construction.

One of the first things learned about materials was that maximum stresses occur in planes farthest from the neutral axis. Both tension and compression are usually present.



■ Fig. 8 – Cycle-Weld assembly fixtures used to fabricate B-26 bomber parts

The efficiency of stressing metal in tension has long been recognized, but trouble begins when attempts are made to stress the compression member. Due to the distribution of loads which result from the application of the monocoque principle, this compression member is only a thin skin. To prevent its premature buckling, it must be stabilized, and that is where adhesives contribute. To be stabilized, or stiffened, its effective thickness must be increased. This is done through the introduction of a low density core material between very thin skins to give a sandwich construction. The result is a greatly increased moment of inertia of the compression member without change of weight.

In the case of any efficient structure, particularly an airplane and certainly in a light car, increasing this resistance to buckling without appreciably adding weight is certainly desirable. Cores, consisting of metal corrugations, cork, balsa wood, and various low density plastics, have been used. These materials are best fastened to the load carrying skin by means of adhesives. Actually, without the continuity of bond that adhesives offer, complete stabilization would still be an assumption in your mechanic book. In the sandwich all of the advantages of adhesives can be realized; the ability to join a wide variety of materials having different properties, the realization of a high degree of structural efficiency with fine economy of materials, clean, smooth surfaces, and ease of fabrication.

As was stated earlier in this paper, the tooling aspects are of considerable importance in design for adhesives. The designer will consider in his work that the parts he is creating must be built economically and efficiently and must conform with production methods. He will also consider that the tool engineer and the mechanic who must design and make the tools must have access to the glue line for the application of heat and pressure.

Consider first the bond line and the general conditions required for a satisfactory joint. Since most of the high strength adhesives are the thermosetting type, two conditions must be fulfilled at the glue line. Pressure must be exerted and simultaneously heat must be applied for a definite time cycle. Pressure is required to ensure complete mating of the faying surfaces, and in most cases has little to do with the chemical reaction of the curing bond. The devices employed should locate the parts properly with reference to each other while applying the necessary pressure. Provisions should be included in the device for follow-up pressure to ensure maintenance of the desired contact despite flashing out or shrinking of the adhesive.

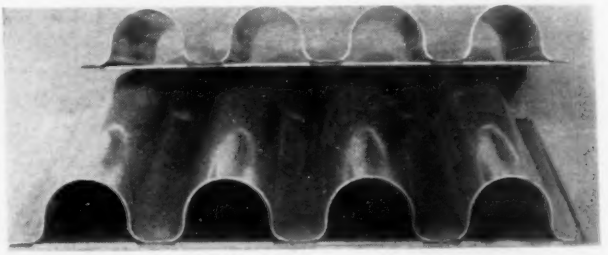
An important factor in consideration of pressure devices is the use of some resilient material interposed between the part being bonded and the platen to distribute pressure uniformly over the entire bonding area as well as to compensate for differences in platen closure and sheet and film thickness. Examples of such materials are chipboard, glass cloth, or galvanized wire covered rubber.

The application of pressure and heat can be accomplished simultaneously through the use of the hydraulic press with flat heated platens, and individual or multiple bench fixtures with toggle or quick-acting clamps and builtin cartridge or strip heaters. The flexible diaphragm press with heated hydraulic fluid and the rubber pressure bag in the autoclave are additional methods employed successfully. (See Fig. 10.)

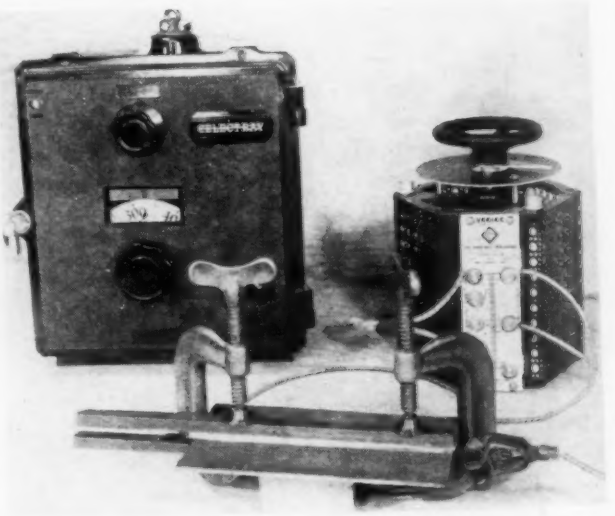
Fitting into continuous production methods are the simple clamping devices and the tunnel oven. Here production rates and schedules are comparable to a paint baking schedule. Pressure is applied by wedges or toggle clamps, and the parts are hung on a conveyor passing through an oven at the proper temperature.

Builtin heaters usually employed are of the enclosed cartridge, strip, and ceramic-bead-protected wire type. Infra red lamps, steam, and hot fluids should be included as sources of heat.

The question of what is done about thermal expansion



■ Fig. 9 – Heavy corrugation assembled by Cycle-Welding



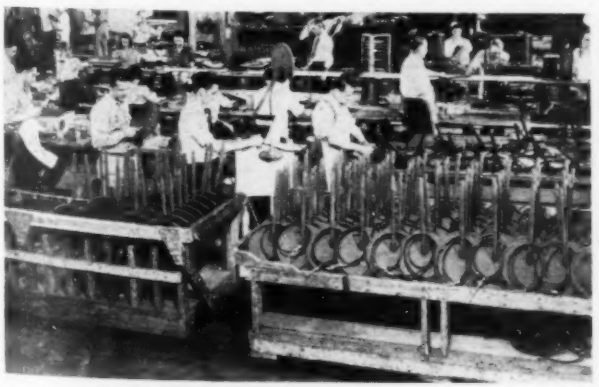
■ Fig. 10 – Typical inductance heating setup using external skin heater

of parts being bonded has often been asked. Where warpage or "oil canning" becomes a factor, and such is occasionally the case on large or irregularly shaped assemblies, certain precautions are necessary to help alleviate the stresses set up. Two simple rules are applied toward this end. First, all portions of the curing die or fixture should be heated to a uniform and constant temperature. Cold or hot spots are bound to produce nonuniform expansion, resulting in either warpage or undue stresses on the bond. Secondly, the parts are allowed to reach maximum expansion before pressure is applied. Parts are placed in the curing fixture in their respective relative positions and the upper platen is just brought short of closure. A short, predetermined time is allowed for the parts to heat up, after which time the press is closed and pressure is applied. In this manner it can be seen that, although not entirely eliminating cured-in stresses, the aforementioned rules will reduce such stresses to a minimum. Fortunately, warpage difficulty is encountered only occasionally.

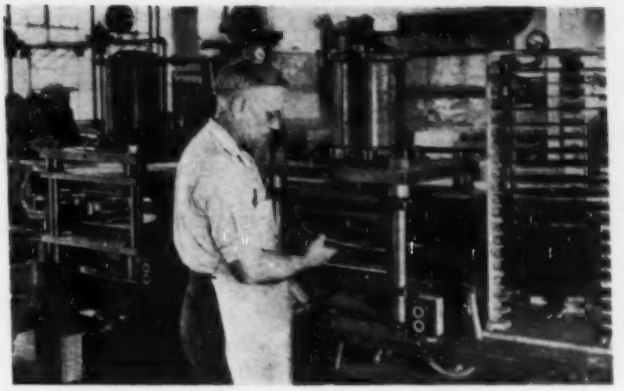
Inspection of the finished bond, although perhaps not important to the designer or engineer, is nonetheless of utmost importance to those interested in quality and reputation of product. At this time there is no positive method of inspection of a cemented joint. Wood gluing presented the same problem some years ago. Years of experience with these glues, however, developed some confidence in their use. Users follow certain mixing instructions, control the process carefully, and apply a few control tests to the wood and glue. Wood being as unpredictable as it is, there are actually more hidden flaws in wood gluing than in metal-to-metal bonding, yet wood gluing has been accepted and used successfully. Structural adhesives are at present in the same category with wood adhesives in that it is possible to control the end product by controlling the process throughout the fabrication cycle. Surfaces to be joined should fit well. Cleaning, application of cement, and precure treatment are frequently inspected for possible deviations from specifications.

Daily sampling of production cements controls the cements being used. All cements must meet minimum strength requirements set for the particular type of cement before it is allowed to enter production. All temperature and pressure ranges required in the process are controlled by automatic devices to control these variables. A constant check is maintained on the complete production cycle.

Final inspection of the finished parts, to a great extent,



▪ Fig. 11 – Search-head for Army mine-detecting equipment being manufactured entirely with adhesives



▪ Fig. 12 – Close-up showing one of cementing operations in manufacture of search-head

is determined by the part. In all cases a certain per cent of production items is destroyed to show quality of bond obtained. Edge contact is checked by the use of feeler gages and the amount of flash is noted. Some materials give visual indication of poor bonding, while others may require sounding with a mallet. In a good bond line the frequency response when tapped is the same over the entire bond. Wherever possible, fixtures and test jigs are used to apply more than the maximum load the part may encounter in use, and all parts are checked therein.

Anticipated inspection of the Cycle-Weld joints will make use of the sonigage, utilizing supersonic vibrations to determine the relative strength of bonds between metal and other materials. A variable frequency oscillator energizes a piezo-electric crystal which is held in contact with the work under test. The driving force causes the metal under the crystal to vibrate, and the reactance between the crystal frequency and the frequency of the metal causes variations in the current used by the oscillator circuit. The voltage applied to the crystal is impressed on the screen of an oscillograph and the fluctuations of the trace are coordinated to indicate very closely the actual bond condition. This method has been demonstrated to reveal not only noncontact areas, but to indicate poor adhesion to unclean surfaces.

We have discussed inspection, tooling, design, and the properties of adhesives. Before putting a part into production one must consider not only these features, but also relative cost of bonding compared to other methods of assembly. A cost and labor survey on several production parts manufactured for aircraft use indicated a direct laborsaving of approximately 40% when compared to ordinary riveting. The saving is substantially increased with the more specialized countersunk riveting. A comparison with ordinary spot welding is not as favorable if the part requires under 120 welds. The direct labor figures become comparable at 120 spots and favorable beyond that figure. It is fair to add that the above data were collected on parts not designed for adhesives but that could be rather easily adapted to adhesives. A few design changes would have materially affected the laborsaving. The time element compares favorably when production methods are employed, such as when multiple bench or continuous oven fixtures are used. In general, it can be said that the adhesive method of assembly is definitely competitive with other methods of attachment. (See Figs. 11 and 12.)

GERMAN AIRCRAFT and

GERMAN aircraft hydraulic systems are analyzed here under the following headings:

1. General review of German practices and policies with regard to design and manufacture.

2. Basic design features of German systems.

3. Representative hydraulic systems as installed in three of the latest German airplanes.

4. Design and operation of hydraulic systems of different planes and manufacturers.

5. German hydraulic developments in relation to American practice.

THIS paper is an attempt to give as completely as possible information that has been obtained on the design of German aircraft hydraulic systems and their component units. An effort was made to accentuate the features and products which appeared to be new in design, unique in construction, or which might have application to current development in the U. S. It is realized that some of the information presented may be well known to many of our designers, but in order to effect a wider distribution and to avoid the possibility of overlooking any important points, details which might be of value have been included.

■ German Practices and Policies

Whereas U. S. tendencies seem to have been to overdesign, the Germans seem to have underdesigned. Their theory apparently was that if trouble developed they could make increases in strength and changes in design as needed. Trouble did develop in a surprising number of cases, sometimes with disastrous results. For instance, their wheels and brakes were designed with such high unit loadings that the brakes had to be relined after 6-10 normal landings. During the Russian campaign, because of insufficient strength of undercarriage and because of the small diameter wheels used, the German planes found extreme difficulty in operating from the soft, rough fields they encountered.

Although many of the airplane companies designed hydraulic units needed in their particular airplane, this practice was not so prevalent as it is in the United States. Most units were designed by one of the big hydraulic firms, that is Elektron, Elma, V.D.M., or Teves, to the specifications of the airplane manufacturers or the Air Ministry. When the airplane manufacturers did design a unit they usually turned it over to one of the big hydraulic firms to manufacture for the entire industry.

Curiously enough, these four big hydraulic companies did little manufacturing of their own products. They served chiefly as design centers, experimental and research laboratories, pilot production plants, and as a source of trained experts to help their licensees to produce their products. Elma produced a larger percentage of its own designed products than any of the other three, but their output was only 3-5% of the total output of Elma products.

Another unusual and interesting point was that the hydraulic firms would obtain the basic hydraulic system requirements for a projected airplane directly from the Air Ministry. They would then design a complete system and attempt to sell it to the airplane builder and to the Air Ministry. Usually, the entire proposal of one of the large companies would be adopted (with many changes over the original design) and that company would furnish all components. By this method any divided responsibility in malfunctioning or poor design was avoided by the airframe manufacturer.

Many of the things we would consider poor design were practiced by the Germans with apparent impunity. For instance, in the Do 335 hydraulic system, in order to provide a continual source of pressure for the brake system and for the cooling shutter control cylinder, a pressure of 450 psi was maintained in the open-center system when all selector valves were in neutral position. This was accomplished by pushing the entire output of both pumps (30 l per min) through an ordinary, spring-loaded relief valve set at 450 psi in the end of the circuit. However, they experienced no trouble with the relief valve or with undue heating of the fluid. In many instances they passed U-cup packings over ports, although they were careful to keep the ports small and to use a large number of them, spaced equally around the diameter of the cylinder.

Electric versus Hydraulic Systems (German Version) – The same arguments of electrical versus hydraulic systems that have raged in the United States seemed to flourish in Germany. Certain aircraft companies tended toward hydraulic actuation and others toward electrical. Dornier, for instance, was basically an electric outfit, the Do 335 being one of their first attempts at hydraulic operation. Even so, all of the controls for the hydraulic valves were electric. Junkers also preferred electrical operation, although lately they have used some hydraulic. The same was true of Focke-Wulf. Messerschmitt and Heinkel, on the other hand, had all-hydraulic actuation; in fact both companies had done considerable design in hydraulic components.

The status of the electrical versus hydraulic controversy had, by the end of the war, reached the point where most of the participants were in substantial agreement that a combination of both methods was the real answer. The consensus of opinion was that the ideal combination was one in which large powers and forces were transmitted

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Grateful acknowledgment is made to Capt. Harry J. Marx, USNR, Bureau of Aeronautics, for the time and effort he devoted to redrawing most of the figures that illustrate this paper.

HYDRAULIC SYSTEMS
Their Components

by R. H. DAVIES
Parker Appliance Co.

hydraulically but were controlled electrically. Units requiring small powers and small forces should be controlled and operated electrically. The Do 335 hydraulic system is a good example of this compromise. Undoubtedly, there would have been more electrical-hydraulic combinations if it were not for the abrupt end of the war and the shortage of copper wire which developed in the past year.

Standardization – There was a great deal of standardization practiced, but standardization was by company and not on an industry-wide basis (except for materials and standard parts, such as nuts, screws, bolts, washers, and packing materials, which were Air Ministry standards). For instance, Teves would standardize on four sizes of relief valves. Elma or V.D.M. might also make four relief valves to the same performance requirements and with the same size port connections for their standard line. But dimensionally and internally the valves for a given capacity manufactured by one company would be totally different from those manufactured by any of the other companies. Similarly, each company, although it used the standard packing material, developed and used its own standard forms for packings. Thus, two or three companies might have a cylinder of the same diameter and yet the packings used would not be interchangeable.

■ Basic Features of German Hydraulic Systems

Types of Systems – The German designers and component manufacturers seemed agreed on open-center systems or designs which were slight modifications of open-center systems (such as the Do 335 system). By using the open-center systems they eliminated the necessity for having their systems under pressure except when actually needed for operation. They also eliminated the weight and other problems which arose with the use of accumulators and unloading valves. The elimination of the accumulator reduced the plane's vulnerability to gunfire. Valves and other accessories with higher leakage rates could be used in their systems because they had eliminated the unloading valve.

In regard to our use of the accumulator and unloading valve, with closed-center systems, in order to reduce the size and power requirements of the hydraulic pump, the German Air Ministry, in conjunction with the major hydraulic manufacturers, had conducted at Rechlin (German equivalent of our Wright Field) some thorough investigations. These had led them to use the open-center system wherever possible. The investigations were part of an overall project not only on hydraulic versus electric but also on pressures, size, weight, and cost of different types of hydraulic systems.

German hydraulic engineers believe that results of these tests indicated that unless the accumulator was of disproportionately large size, little or no gain in overall speed of operation was realized. It was their finding that, although the accumulator speeded up the operation in the early part of the cycle (when the load was usually light), in the latter part of the cycle (where the load was usually heavy) the pump wasted much of its output in recharging the accumulator. Sometimes the accumulator caused such rapid operation in the early part of the cycle that restrictor valves or constant-flow valves had to be used to avoid undue acceleration forces. In the few cases where this was not true the addition of a slightly higher capacity pump would get the desired results with a considerable weight saving over the accumulator and unloading valve. The additional power requirement of the larger pump was of no consequence, since it was only required when operating the unit requiring the maximum power.

In regard to pressures of systems, the 1500-psi systems were pretty well standardized in most of the modern planes. Some of the newer ships had gone to 2250-psi systems. The accessory manufacturers had been instructed to have ready units capable of handling 3000 psi. Most of them had been able to adapt their standard products with minor modifications, but to anyone's knowledge no airplane was proposed with a 3000-psi system. The Air Ministry was fully cognizant of the advantages of higher pressure systems. They had circulated to airplane and accessory manufacturers detailed studies of weight savings to be made by going to pressures of 3000 psi and 4500 psi.

Packings and Oil – Packings, although not standard from one manufacturer to the other, were mostly U cups, half U cups, or modifications of these designs. Only in one instance (V.D.M.) were any O rings seen, and then only in a few units. Many packings used were specially molded

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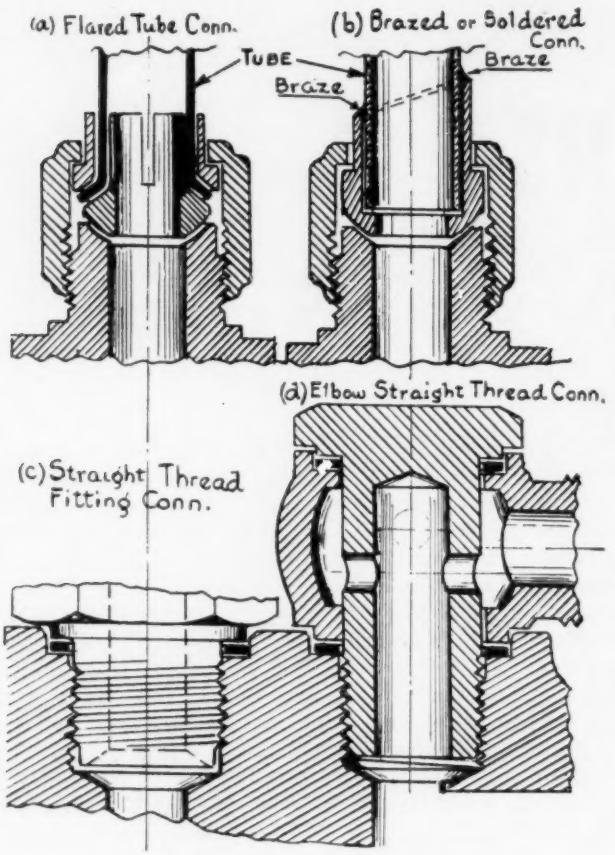


■ Fig. 1 – German tube fittings and straight thread connections

designs to be incorporated with specially designed retaining devices. Most packings used were molded by either C. Freudenberg Simmerwerke at Weinheim (between Heidelberg and Darmstatt) or at the Continental Co. of Hanover.

One interesting point was brought out in regard to packing materials versus hydraulic oils. Apparently the Germans had a very satisfactory packing material from the standpoint of high tensile strength, good cold-flow characteristics, wear resistance, sealing properties, and high- and low-temperature characteristics. When it was discovered that new hydraulic fluid designed for flat viscosity properties had a deleterious effect on this packing material, the entire problem was given to I. G. Farben, since they made both the oil and the packing material. Farben chose to change the oil and keep the packing material the same.

Fittings and Flexible Hose – Although previously a good deal of steel tubing and brazed-on fittings had been used the latest tendency was to use only aluminum tubing and flared fittings, except in the engine compartment, where steel tubing and brazed-on fittings were still used exclusively. A great deal of flexible hose was used by the Germans where we would have used rigid tubing. Straight thread make-up was used to the exclusion of all pipe thread connections. Usually, straight fittings were used to connect to accessory bosses. Where elbows or tees were needed, a banjo-type fitting similar to our automotive brake fitting was used. The tube fittings used were all one type, and had an external thread and inverted cone similar to our AN-800 series fittings. (See Fig. 1.) Where a flared joint was used an olive with a thin split sleeve extending up inside the tube was used as an adaptor. Where a brazed or soldered joint was used, the tube was brazed or soldered into a cone-type sleeve, much like our AN-800 series sleeve. A standard nut, much like our AN-818, was used with either connection. With the flared tube connection a sleeve much like our AN-819 sleeve was used. With the straight thread connection in the accessory bosses a metal gasket completely constrained was used. Fig. 1 shows the different types of fittings and straight thread make-ups used.

All flexible hose was without a protecting covering of rubber outside the braid. Hose fittings were by Argus and were of the detachable end type, practically identical to our AN standard hose ends by Aeroquip. Argus disconnects similar to our Aeroquip were used at the engine firewall, brake connections, and for ground test connections.

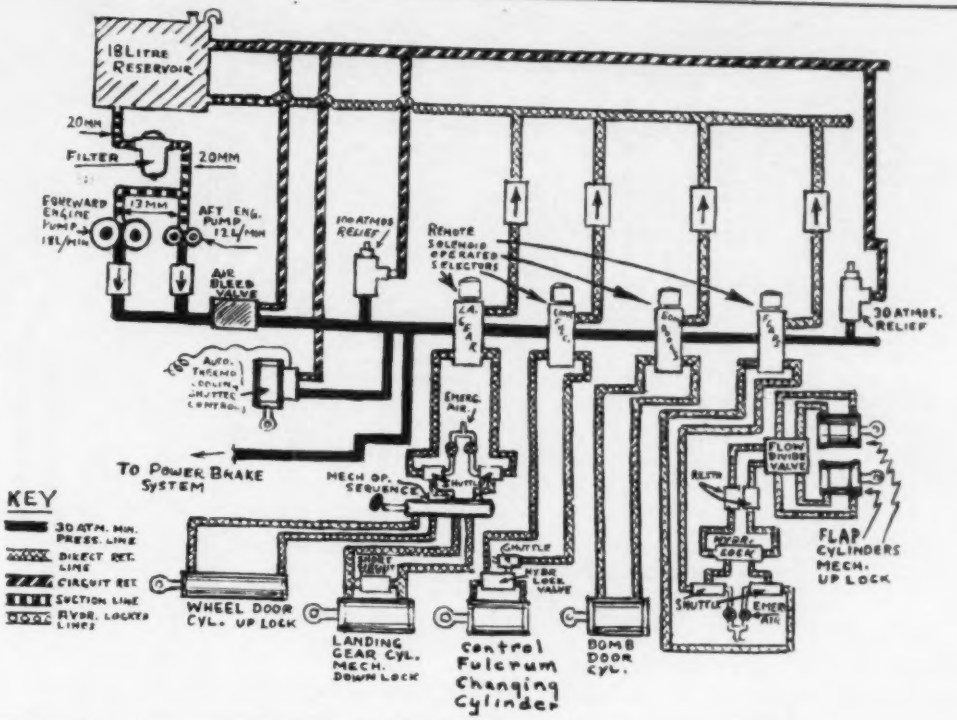
■ Representative German Hydraulic Systems

The hydraulic systems of three of the latest German airplanes were chosen for study as being representative of the best German practice. The three airplanes are the Do 335, the Me 262, and the Me 410.

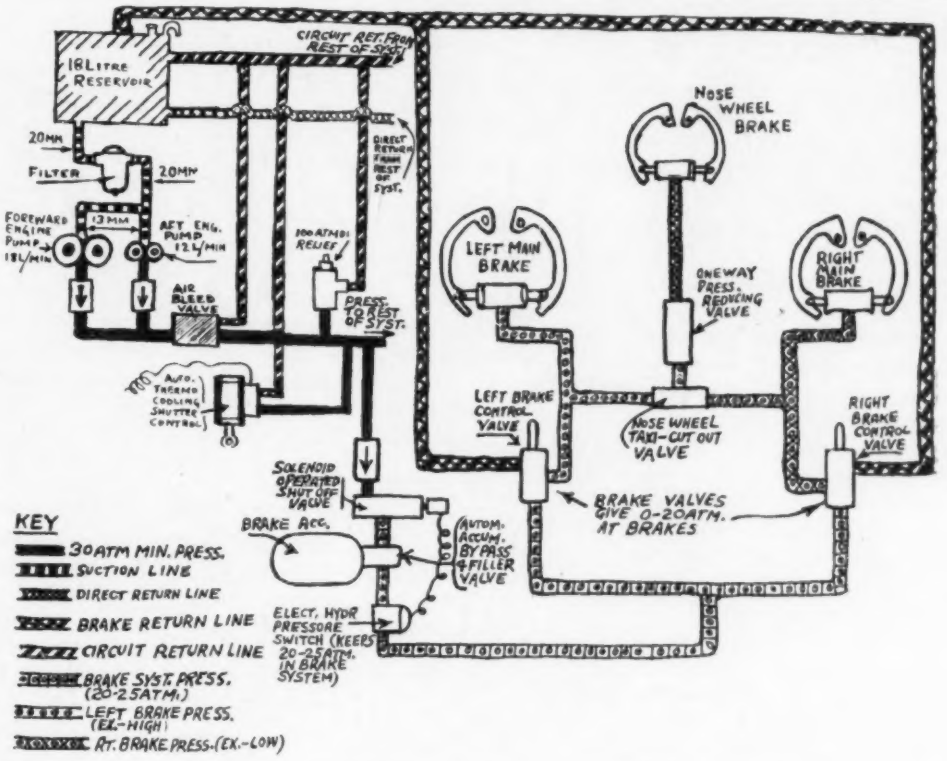
Do 335 Hydraulic System – The Do 335 is a fast, unique fighter with two engines mounted in tandem; one driving a tractor propeller in the nose and the other a pusher propeller in the tail.

The hydraulic system is a semi-open-center system operating at about 1200 psi. (See Fig. 2A.) There are two Barmag gear pumps hooked up in parallel. Both are engine driven, one by the forward engine (18 l per min), the other by the after engine (12 l per min). The reservoir has a capacity of 18 l. On the suction side of the pumps there are located two thin metal edge type of filters, which filter 100% of the fluid before it gets to the pump intake. These filters are fitted with a bottom section which is removable for cleaning. Just downstream of each pump is an air bleed valve whose purpose is to unload the pump and bypass its flow to the reservoir if it picks up any appreciable amount of air. This keeps the pump from becoming airbound and thus burning itself up. This process of unloading is automatic. When the pump has cleared itself the air bleed valve will also automatically return to its normal position and again permit the pump to supply fluid under pressure to the system.

In the main part of the system there is no accumulator. A main system relief valve is set at 1500 psi. On the main pressure line between the pumps and the first selector valve, two lines are taken off; the first goes to the cooling shutter control cylinder, and the second is for the brake system. Each of these has a separate return line to the reservoir. As in any normal open-center system, the selector valves are in series. However, the unusual feature of the system is that 450 psi is maintained in the pressure line even when no units are being operated. This is done by a second main system relief valve, installed in the pressure line downstream from the selector valve. Although this procedure means that when no units are operating the output of both pumps (30 l per min) is put through this second relief valve, the Dornier hydraulic engineers claim no undue heating is experienced in the system. The purpose of this hookup is to provide always a minimum pressure of 450 psi for operation of the cooling shutter cylinder and the brake system. The brake and cooling shutter



▪ Fig. 2A – Do 335 hydraulic system

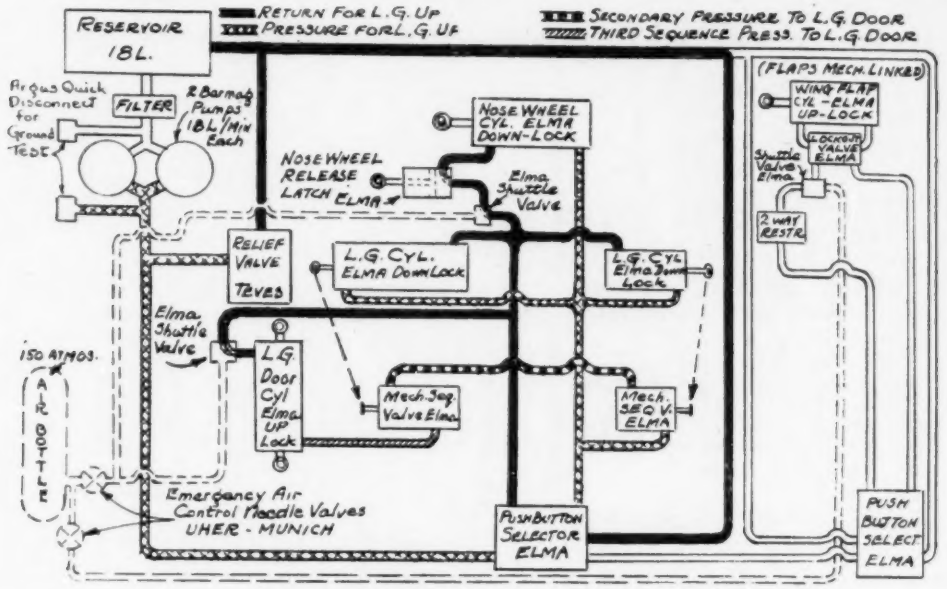


▪ Fig. 2B – Do 335 power brake system

cylinders are in parallel with each other and with the rest of the system.

In order to have available full operating pressure for actuating the landing flaps, landing gear, and control fulcrum changing cylinder, the selector valves are designed so that when they are operating the return side connects with another return line directly, without having to go through the 450-psi relief valve. Also, when the selector valve is in neutral both cylinder lines are connected to this direct return line via check valves to prevent any pressure from inadvertently getting to the cylinder and unlocking it.

No unloading valve is used in the circuit. No hand pump is provided for emergency operation. Air at 950 psi is used for emergency operation of landing gear, landing flaps, and changing of control fulcrum. Ordinary needle-type valves manufactured by Uher in Munich are used for


RESERVOIR 18L.
RETURN FOR L.G. UP
PRESSURE FOR L.G. UP
SECONDARY PRESSURE TO L.G. DOOR
THIRD SEQUENCE PRESS. TO L.G. DOOR
(FLAPS MECH. LINKED)
Argus Quick Disconnect for Ground Test
FILTER
2 Barnes Pumps 18L/MIN Each
NOSE WHEEL CYL. ELMA DOWN-LOCK
NOSE WHEEL RELEASE LATCH ELMA
Elma Shuttle Valve
WING FLAP CYL - ELMA UP-LOCK
LOCKING VALVE ELMA
Shuttle Valve Elma
2 WAY RESTR.
RELIEF VALVE TRVES
L.G. CYL. ELMA DOWN LOCK
L.G. CYL. Elma Down Lock
Elma Shuttle Valve
ISO ATMOS.
AIR BOTTLE
L.G. Door Cyl Elma UP Lock
Mech. Seq. Valve Elma
Mech. SEQ V. ELMA
Emergency Air Control Needle Valves UHER - MUNICH
PUSHBUTTON SELECTOR ELMA
PUSH BUTTON SELECT ELMA

■ Fig. 3 – Me 262 hydraulic system

controlling the air. Guns are also charged by air. The emergency air is throttled so that at the landing gear cylinders there is 600 psi, at the flap cylinders there is 900 psi, and at the control fulcrum changing cylinder there is 450 psi. The bombdoors, brakes, and other units are not provided with emergency air operation.

The following equipment is hydraulically operated on the Do 335: (a) landing gear, (b) landing gear doors, (c) landing flaps, (d) bombdoors, (e) flight control fulcrum changing cylinder, (f) brakes, and (g) cooling cylinders.

On previous Dornier aircraft electric operation had been used exclusively. The change to hydraulic operation on this model was done in most cases to reduce weight and simplify the installation. However, some units Dornier would have preferred to operate electrically but they were forced to hydraulic operation by the Air Ministry standardization program and the shortage of copper wire.

Equipment – (a) Brakes: The Do 335 brakes were of the shoe type and were acknowledged to be inferior to our disc-type brake. (See Fig. 2B.) The Germans have a great admiration for our disc-type brake, and wanted to use them on the Do 335. They were prohibited from doing so by the Air Ministry standardization program, but just before they ceased operations they had received permission to change to a new disc-type brake developed by Elektron.

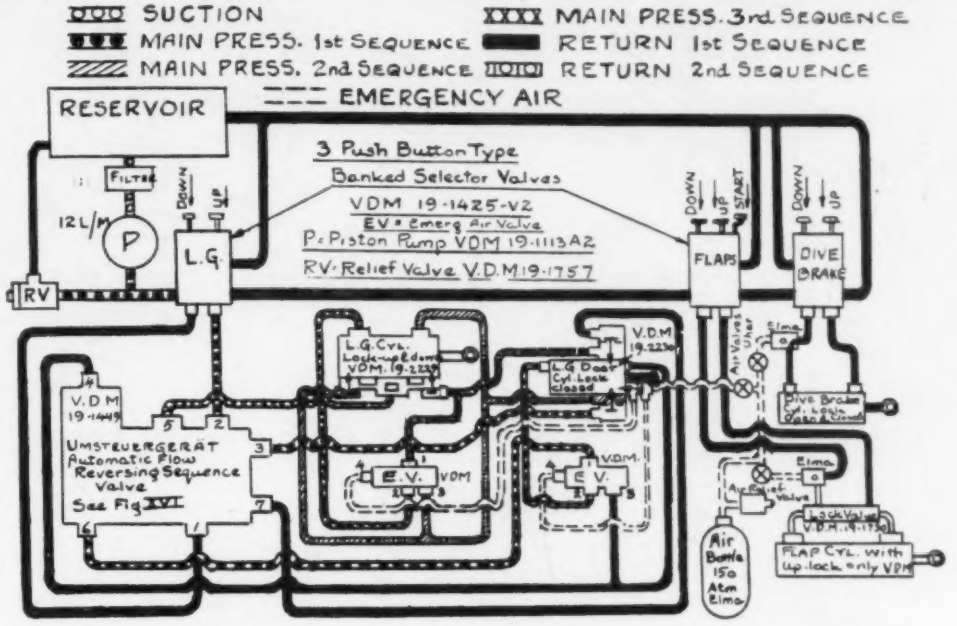
All rudder pedals were standardized and the power brake control valve was assembled as a part of the pedals, flexible hose being used to connect to the brake valve. The valve itself (also made by Elektron) is unique in that its designers have accomplished very simply a balanced hydraulic construction with an incorporated hydraulic feel piston, which operates directly against the pedal and is independent of any springs or breathing action of the valve, thus leaving nothing but direct hydraulic feel against the pedal.

In the brake circuit was a bag-type accumulator, the outer bottle of which was the same as used for oxygen bottles. This accumulator is teed into the brake line with an automatic bypass valve which will automatically bypass the accumulator in case it is shot away.

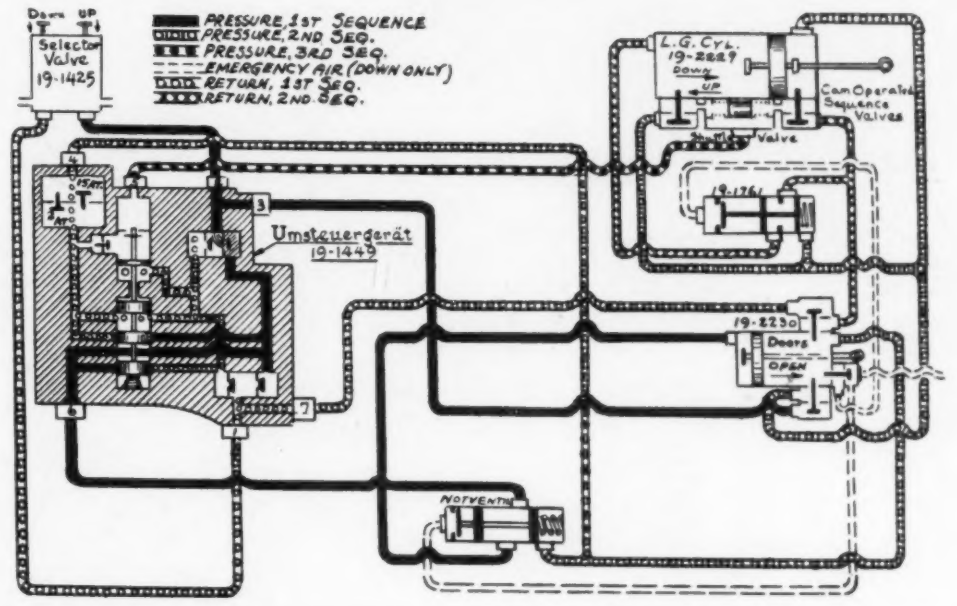
All three wheels of the tricycle landing gear are braked. However, the nose wheel is tied into the system via a shuttle valve, which permits pressure to go to the nose wheel only when equal or nearly equal braking pressure is being applied to both main wheels. This is to preclude the nose-wheel brake from acting during taxiing. Also in the nose-wheel brake line was a pressure reducing valve which kept the nose-wheel pressure about 100 psi lower than any pressure applied to the main brakes. The main braking pressure was 300 psi maximum.

(b) Actuating cylinders: The actuating cylinders were similar to ours in general workmanship, materials, and finishes except that U-cup packings were used exclusively. The landing gear cylinders were equipped with builtin ball-type mechanical locks, hydraulically operated for the down position. The flap cylinders were provided with a similar lock for the up position. Since all selector valves were solenoid operated automatically return-to-neutral type, each cylinder was provided with bosses at either end to fasten to a standard type of electrical limit switch to throw the selector valve into neutral when the cylinder reached the end of its stroke. Most cylinders were manufactured by V.D.M. in Gros Anheim or by Elma in Waiblingen.

(c) Cooling shutter control unit: This is an unusual device manufactured by V.D.M. It consists of a thermostatic feeler in the cooling system which acts through a hydraulic servo to regulate the pressure on the cooling flap cylinder. It is claimed to be very satisfactory and very accurate in its temperature control. The entire device (with the exception of the feeler) is one unit. A combination mechanical and hydraulic device which consists of a short circuiting valve and a mechanical push-pull control is provided in case the pilot wants to override the thermostatic control. This first short circuits the hydraulic cylinder and then provides a push-pull mechanical control for the cooling shutters. In some of the planes a hydraulic selector valve is used which cuts out the thermostatic automatic control and substitutes hydraulic operation controlled by the pilot. The former device is manufactured by Telekin of Berlin and the latter is manufactured by Elma at Waiblingen.



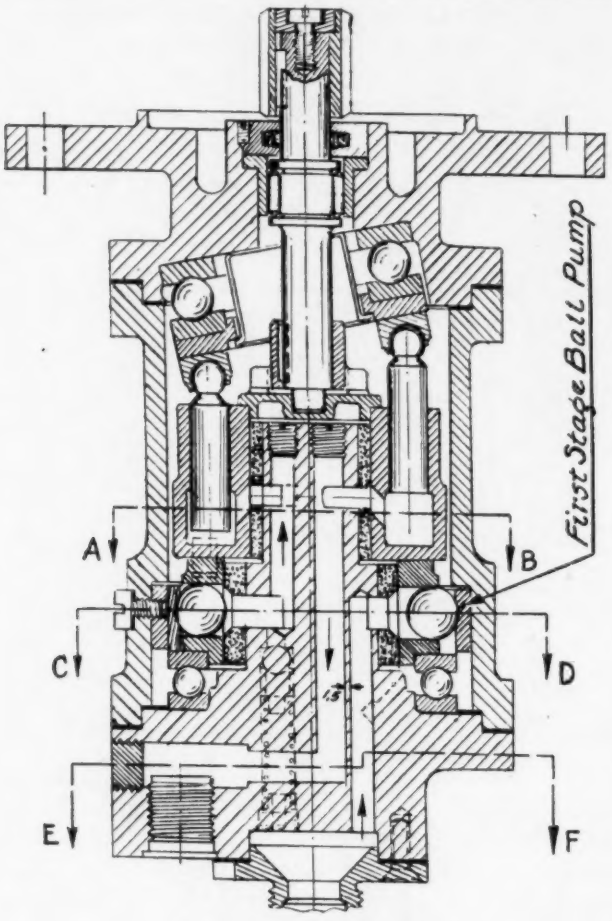
▪ Fig. 4A – Me 410 hydraulic system showing pressure sequence for gear-up operation



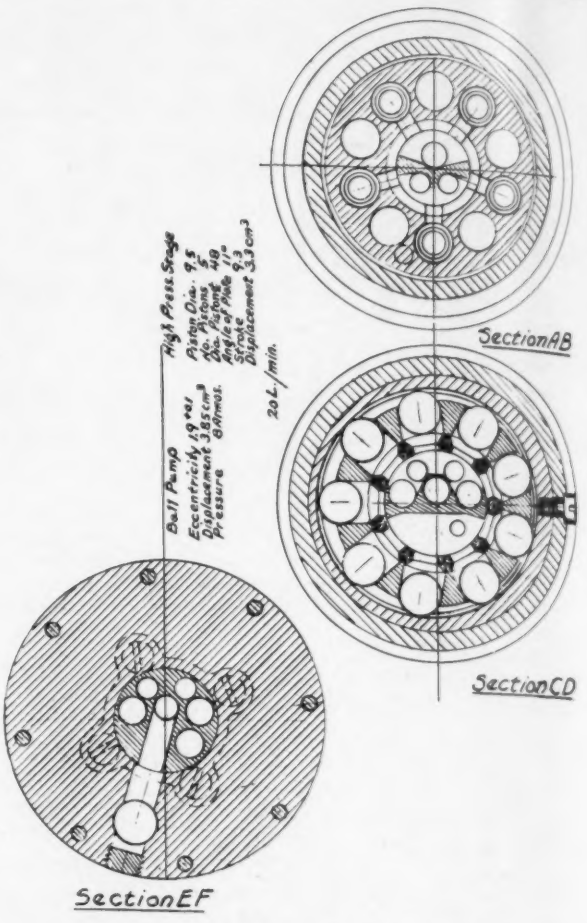
▪ Fig. 4B – Me 410 landing gear circuit with schematic diagrams of valves

(d) Selector valves: The selector valves are all solenoid operated automatic return-to-neutral type valves. They are automatically returned to neutral when the cylinder reaches the end of its stroke by the solenoid actuated by limit switches on the actuating cylinders. The valves are of the slide, lapped-fit type having five ports. No. 1 port is pressure in and No. 2 port is pressure out when valve is in neutral. No. 3 port and No. 4 port, which lead to the two ends of the actuating cylinder, are both connected to the direct return line via port No. 5. A check valve is also included in the direct return line leading from port No. 5. This hookup ensures that no pressure can inadvertently get to either end of the cylinder and unlock it when the selector is in neutral. No. 5 port is to the direct return line and is open when No. 2 port is off and pressure is going out of either No. 3 or No. 4 port, when the valve is not in neutral. When the valve is in neutral No. 5 port is open to both No. 3 and No. 4 ports.

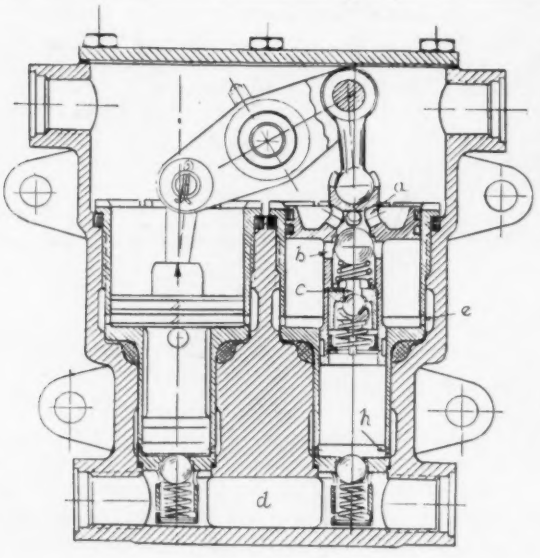
(e) Flight control fulcrum changing device: An unusual way of obtaining boost in the controls for high-speed performance is obtained by moving the fulcrum point of the control system by means of an hydraulic cylinder. Thus, for landing at very low speeds there is a position where force exerted is low and movement of the control surface is great. For normal cruising there is an intermediate position where force exerted is higher and movement less. Then for every high-speed work, such as diving, there is a third position where very high forces can be exerted but very little movement of the control surfaces results. This change of leverage of the controls is limited to the ailerons and the elevators and does not affect the rudder. Recently, it was decided to eliminate one of the three positions and

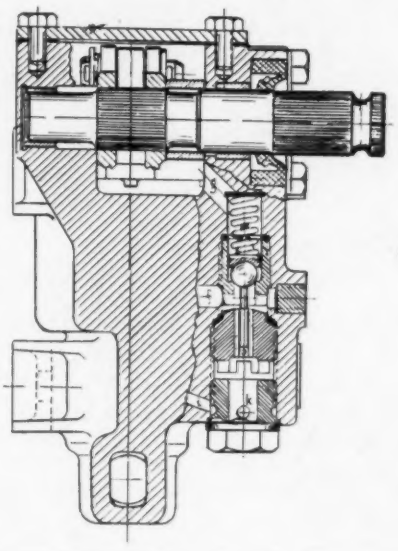


■ Fig. 5A – Teves hydraulic pump



■ Fig. 5B – Sections through Teves hydraulic pump

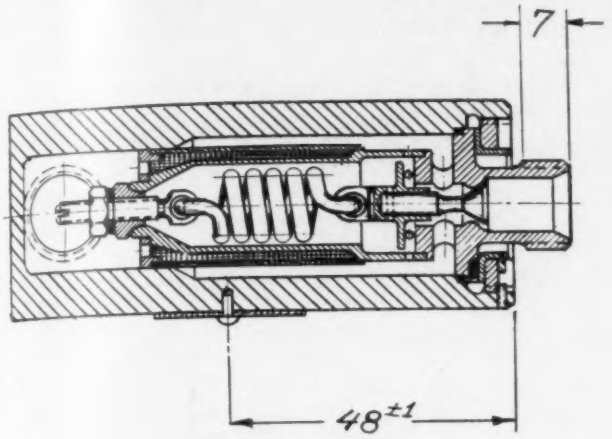




■ Fig. 5C – Teves hand pump

have only the high-force and low-force positions.

(f) Miscellaneous: The flap cylinders and the control fulcrum cylinder are provided with hydraulic lockout valves which have thermal relief valves incorporated in them. Only three positions of the landing flap are available, namely, flap up, landing position, and take-off position. Restrictor valves are used in the flap circuit (both hydraulic and emergency air) to ensure proper speed of operation. An hydraulic flow divider said to be accurate to within 5%, with pressure requirements varying from 150 psi in one flap cylinder to 900 psi in the other, is used in the flap circuit. This has a builtin thermal relief valve



■ Fig. 6 – V.D.M. relief valve

as well as a device for exactly synchronizing the flaps at either end of the flap cylinder stroke. This flow divider was designed by Junkers and was being built at Elma.

Since the landing gear will fall into its down position by gravity all landing gear actuating cylinders are equipped with automatic short-circuit valves.

Necessary check valves of conventional design are used. These are not equal in performance to American AN check valves and would not meet our AN specification. The wheel door cylinders are connected with the retracting cylinder circuit by means of hydraulic sequence valves which, on the down operation, do not permit the wheel cylinders to function until the doors are opened. On up operation they do not permit the wheel doors to close until the wheels are fully retracted.

Me 262 Hydraulic System – This airplane is a very fast twin-engine jet-propelled fighter bomber. The Me 262 has a full open-center type of system using mostly Elma equipment. (See Fig. 3.) The selector valves are of the manual push-button automatic return-to-neutral type and are used for operating the landing gear and wing flaps. The pumps are Barmag 18 l per min gear pumps with line type metal edge filters on the suction side. Two pumps in parallel are used in the latest version, one driven by each turbine.

The main system relief valve is by Teves of Frankfurt. It is a straight spring-loaded relief valve set for 1700 psi.

The landing gear cylinders are by Elma and are equipped with builtin mechanical down locks which are hydraulically released. The landing gear door cylinder is by Elma and has incorporated in it mechanical locks for the closed position of the door, these locks being also hydraulically operated.

The wing flap cylinder has a hydraulic lockout valve used with it; the flaps being mechanically linked, need only one cylinder. The cylinder and lockout valve are made by Elma.

Air-operated shuttle valves which are spring-loaded in the normal position are used on the flaps for putting flaps down only and are used on the landing gear door cylinder for opening the door only. One is also used on the nose-wheel cylinder for lowering the nose-wheel gear. Since the main gear will drop and lock by gravity when the main doors are open, no emergency air is provided for their operation.

For closing the landing gear doors two mechanically operated sequence valves are used in series, each operated by one of the main wheels. Thus, the main door cylinder cannot start to close the main doors until both main wheels are fully retracted. The mechanically operated sequence valves on the landing gear door (closed) circuit also allow free flow when the wheels are being lowered.

The nose-wheel door is operated by a mechanical linkage connected with the nose wheel.

For locking the nose wheel up a spring-loaded latch or hook is used. Before the nose wheel can be lowered, that is, before pressure can be applied to the nose-wheel cylinder to lower, the locking latch must be withdrawn hydraulically and the wheel released. Only when this is done can pressure reach the nose-wheel cylinder. This operation is accomplished by the Elma control cylinder valve. During return flow, that is, when the gear is being raised, the valve allows free flow.

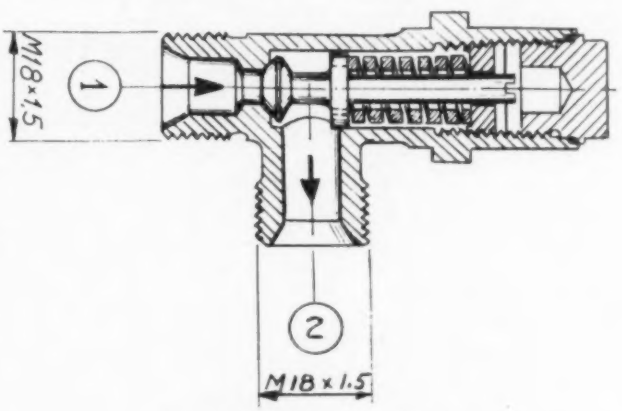
The emergency air system uses a 2200-psi air bottle and two Elma push-button on-and-off valves, one for flap down and the other for landing gear down. Earlier models used Uher (Munich) needle valves in place of Elma push-button valves for emergency air operation. Necessary check valves are used in the pump circuits so either can be used in case of failure of the other.

Me 410 Hydraulic System – The Me 410 was a fast, twin-engine general-purpose fighter-bomber. The Me 410 system was quite different in a great many respects from any other German aircraft hydraulic system. It was almost completely furnished with equipment by V.D.M. of Gros Anheim, instead of by Elma, as with most of the other German airplanes. Fig. 4A shows the entire system and Fig. 4B, the landing gear circuit.

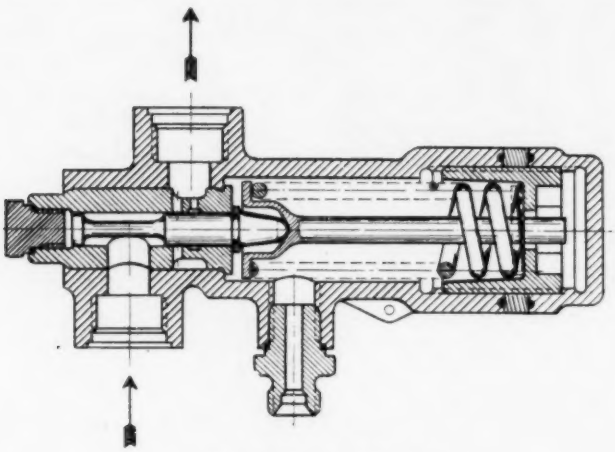
There were two 12 l per min V.D.M. pumps used in parallel, rated at 100 kg per sq cm pressure. A metal edge line filter was used in the suction line between the pump and the 12-l reservoir. A V.D.M. relief valve was used.

The system is an open-center type with separate return lines from each selector valve. The landing gear, wing flaps, and dive brakes are operated hydraulically.

The selector valve used is a stack of three valves mounted together, each one controlling one of the hydraulic units (landing gear, flaps, dive brakes). The valves are of the lapped-fit slide type.



■ Fig. 7 – Elma pressure relief valve



■ Fig. 8 – Teves relief valve

Emergency air at 2200 psi is provided for emergency operation, as follows: (a) lowering the landing gear, (b) extending the dive brakes, and (c) lowering the landing flaps.

The selector valves are so designed that when they are in the neutral position, both cylinder lines are open to the direct-return line (via a check valve to prevent any pressure buildup in the return line from getting to the cylinder and unlocking it). They automatically return to neutral at the end of the piston stroke. This is accomplished not by using the buildup of pressure which occurs at this point, but by using the principle that at the end of the piston stroke the flow of oil stops. The operation of the selector valve may be reversed at any time during the stroke of the piston by depressing the opposite button.

In order to stop the flaps at some intermediate point the flap selector is provided with a third button marked "start." This button must be held down to lower the flaps. The valve returns to neutral at any time pressure on this start button is released. If the "ausbutton" is depressed to lower the flaps it will stay in position automatically until full flap is reached.

The landing gear cylinders (V.D.M.) have incorporated in them a segment-type mechanical locking device for locking the gear down and up. This lock is better than Elma's, although it is more expensive to build. The lock is hydraulically released when pressure is applied against the piston.

The landing gear cylinder also has two mechanically operated sequence hydraulic valves, one built into each end and actuated by the piston.

The landing gear door cylinder (V.D.M.) has incorporated the same type of lock but for the closed position only. It also has two mechanically operated hydraulic sequence valves, both actuated by the piston when it is in the door-open position. At this same end of the cylinder is another mechanically operated air sequence valve for the emergency air circuit.

The landing gear cylinder also has attached to it a spring-centered shuttle valve which takes pressure from the lower chamber of either sequence valve and leads it to the operating piston of the automatic flow-reversing sequence valve which is also included in this circuit.

The flap cylinders have no mechanical locks, but with them are used hydraulic lock-out valves so that they are hydraulically locked whenever the selector valve is in neutral. Restrictor valves are used in the flap circuit to reduce the speed of operation. The flap cylinder has electric switches operated by the piston to indicate when the flap is up or down.

The flap and dive brake circuits were conventional, but the landing gear circuit was unusual and used several unique units. This was done because it was necessary on the 410 to have the landing gear doors closed both when the gear was retracted and when it was extended in order to cut air resistance on take-off. The hydraulic operation of the landing gear circuit is explained later in paper in the section devoted to special units.

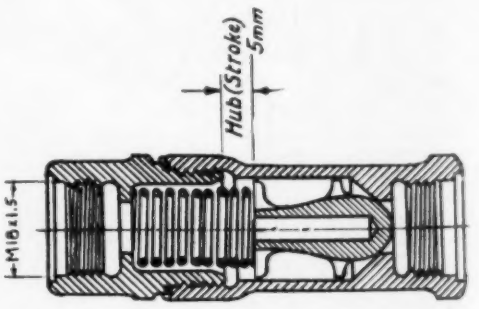
The emergency air system, when operated, allows pressure in the top chamber of the air sequence valve on the door cylinder to go to the door open side of the door cylinder after it pushes over the door cylinder emergency valve piston, opening the other side of the door cylinder to the return line. When the door cylinder reaches the end of its travel it opens this emergency air sequence valve on the door cylinder, allowing the air pressure in the top chamber of this valve to go to the side of the landing gear piston, which lowers the landing gear. However, it must go via the landing gear emergency valve, pushing it over and opening the other side of the landing gear cylinder to the return line. Thus, the doors are opened and the landing gear lowered by emergency air. No provision is made for closing the doors after the gear is lowered or for raising the gear by emergency air.

Valves used for controlling the emergency air are either Uher needle valves or Elma push-button on-and-off valves similar to our manual oil dilution valve. The air system is provided with an Elma soft-seated poppet-type relief valve seating at 2200 psi.

■ Design and Operation of Hydraulic Units

The hydraulic accessory units described below were selected from a large number of German designs. They are believed to be generally the most popular and most modern types in each particular class of equipment. Most of the units are products of the three major aircraft hydraulic manufacturers, namely, Teves, Elma, and V.D.M.

Hydraulic Pumps – Most German pumps in use were of the conventional-gear type Barmag pumps. Another type by Teves showing promise at the end of the war (Figs. 5A, 5B) is a conventional piston pump with rotating pistons, except that it incorporates a first-stage pump made up of close fitting balls which operate as radial pis-



■ Fig. 9 – Elma check valve

tons. No springs are, therefore, needed on the main pistons. No springs are needed on the balls (first-stage pistons), because centrifugal force keeps them out for the suction stroke.

Hand Pumps – The double piston Teves hand pump incorporates a number of features which are intended to give maximum performance, even at the expense of simplicity. (See Fig. 5C.) The pump is designed to pump oil at pressures up to 1500 psi. In order to reduce handle loads at high pressures, stepped pistons with an area ratio of about 4 to 1 are used. When the delivery pressure reaches a preset level the large end of the piston is bypassed and the small end of the piston supplies the high-pressure ranges. As a consequence the pump handle load is maintained approximately constant over the whole range of operating pressures. The bypass feature is entirely automatic and utilizes a pressure loaded pin to open a bypass check valve and unload the large end of the pistons.

Relief Valves – This is a unique design relief valve for direct in-line installation with a relief port on the top of the valve. (See Fig. 6.)

The unusual feature of the valve is the use of a tension spring pulling a loosely guided poppet into its seat. The seat and the poppet are both metal with differential angles for line contact seating.

The tension spring construction was used in an attempt to ensure true seating, minimum friction, and therefore better overall performance. Outside of the difficulties which could be expected with tension spring design, the idea seems good. The performance appears to justify the radical design. The construction is cheap and simple. At 1650 psi the valve cracks, giving a flow of 16 l per min at 1725 psi and closes to 100 cc per min leakage at 1650 psi.

Fig. 7 shows a simple poppet type of direct spring load design. A square spring is used, and the poppet is guided in the bore with another guide where the poppet extension goes through the top plate (which retains the spring). This top plate (since it is screwed in) acts as a means of adjusting the relief pressure. A gasket cap is used and seals in the oil as well as covering the adjusting mechanism. A screw driver slot is provided in the poppet extension for lapping it in for a tight seal on the seat.

Fig. 8 shows one of a series of conventional slide type of relief valves. The most interesting point in their design is the fact that with the valve set to crack at just above 100 atmospheres, at 100 atmospheres only 5 drops per min is allowed.

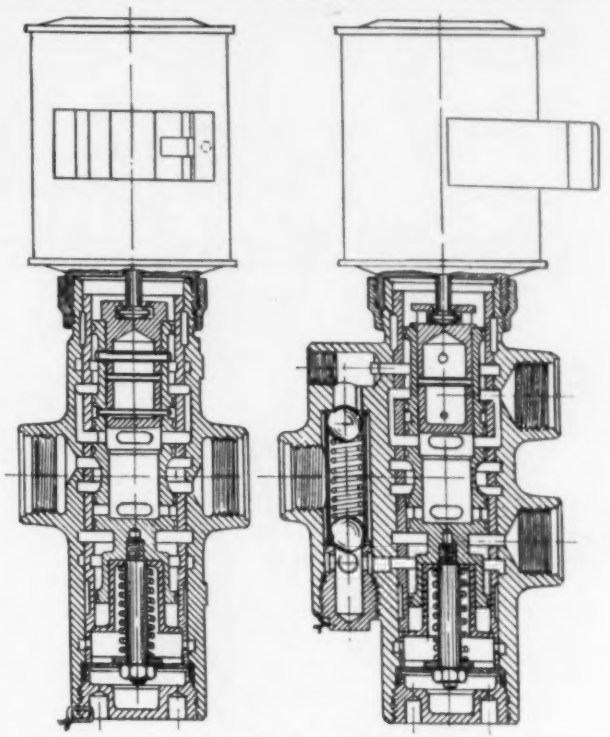
The valve shown is rated for 30 l per min flow, being a late development incorporating a dashpot chamber under the piston for damping vibration.

This line of Teves relief valves is acknowledged throughout the industry as being the best of German design.

Check Valves – Fig. 9 shows a conventional type of check valve. Of special interest is the streamlining of the poppet. Evidently this was done to reduce pressure drop. It achieves this in high rates of flow, where the poppet is back against the stop, but in low rates of flow it actually increases the pressure drop, since the orifice between the poppet and the seat is reduced.

Selector Valves – The solenoid selector valve (Fig. 10) and manual selector valve (Fig. 11) are balanced slide type of selector valves.

Fig. 10 has the same valve body as Fig. 11 but is solenoid instead of manually operated. In the solenoids, the wiring



■ Fig. 10 – Elektron solenoid valve

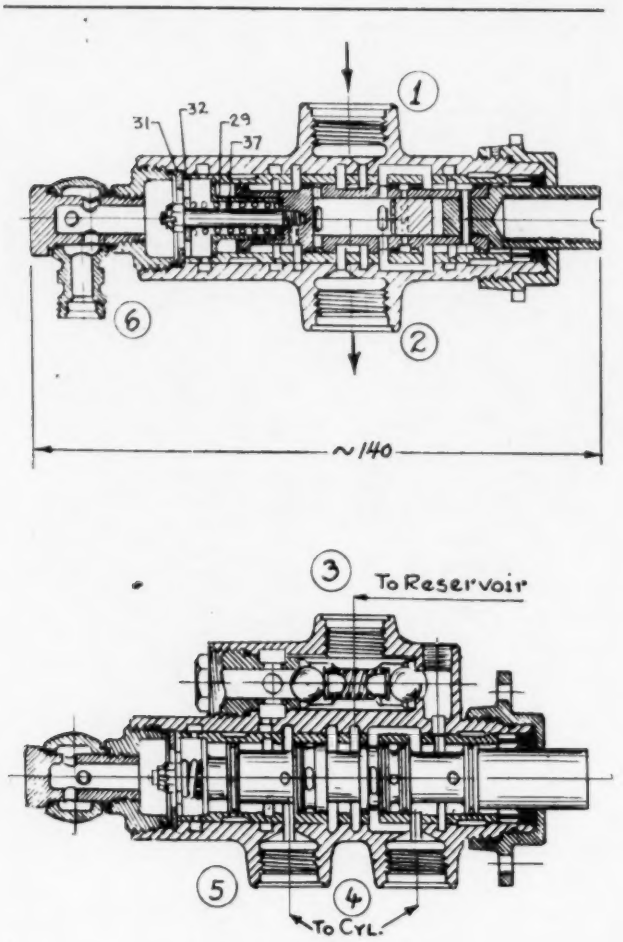
was sealed so that oil, even though it got into the plunger chamber, could not get to the wiring.

The valve has three positions: (a) cylinder down, (b) cylinder up, and (c) neutral.

The valve was designed for automatic return to neutral when the piston reached the end of its travel. The return to neutral from either position was accomplished by the use of only one spring. The valve body itself was standard and could be adapted to manual as well as solenoid operation (or even remote hydraulic operation, although this method was not used).

For solenoid operation, the valve was permitted to return to neutral by means of limit switches in both ends of the cylinder actuated by the piston. For manual operation the valve was returned to neutral hydraulically by means of an attachment which went on the end of the valve, where the solenoid was mounted in the electric version. This attachment consisted of a spring loaded detent mechanism for holding the slide in either acting position against the action of the centering spring. For pulling the detent pin out (thus allowing the centering spring to bring the valve to neutral at the end of the piston stroke), a small, spring loaded hydraulic piston was provided. When the piston reached the end of its stroke, pressure was built up high enough against the small piston to overcome the spring loading, thus pulling out the detent pin and allowing the centering spring to operate and return the valve to neutral.

One spring in the valve did the centering, as follows: When the valve slide was pushed to the left (see Fig. 11), guide 32 was against stop 31, so guide 29 would be moved to the left by valve slide, thus compressing spring 37. When valve slide was moved to the right (from neutral position), guide 29 was against the stop in the steel liner



■ Fig. 11 – Elma slide valve

sleeve, so guide 32 was pulled to the right (by pin in valve slide), thus also compressing the spring 37.

When the valve is in neutral position, both the cylinder ports of the valve (ports 3 and 4) are open to the direct return line to the reservoir (port 5) via two check valves. These check valves are provided as a safety measure, so that any pressure building up in the return line could not get back to the actuating cylinder and inadvertently unlock it.

When the valve is in the neutral position, ports 1 and 2 are connected. Pressure fluid goes into port 1 and out of port 2 to the rest of the system. When the valve is in the operating position, port 1 is connected to either port 3 or port 4, while port 2 is shut off, and the direct return to the reservoir (port 5) is connected to pressure port 1.

In Fig. 11 the banjo connection on the left is needed to allow leakage to go to the reservoir, so that no pressure can build up on the left end of the valve slide.

Push-button hydraulic selector valve: Fig. 12 shows a balanced, double slide type of selector valve used on the Me 262. German pilots liked this valve particularly well because it has a separate button for cylinder down and cylinder up.

In the neutral position, both push buttons are up. Port 1 (pressure in) is connected with port 2 (pressure out to rest of system). Ports 3 and 4 (cylinder ports) are connected to port 5 (direct return to reservoir). For cylinder down position push-button 1 is depressed. The pressure required is light since both slides are balanced hydraulically and the detent friction is the only load to be overcome. In this position ports 1 and 3 are connected, directing the pressure oil to the cylinder down port. Ports 4 and 5 are connected, allowing the return oil to go directly to the reservoir. Port 2 is shut off. Just beyond the pressure port is a poppet type of relief valve set for 50 to 100 atmospheres. The relief side of this valve is connected to the ends of both slides. Thus, when the cylinder reaches the end of its stroke the pressure builds up, opening the relief valve, thus allowing a small pressure buildup on the bottom of the slides. This is sufficient to overcome the detent and return the depressed slide to its neutral position.

Solenoid pilot type of selector valve: Fig. 13 shows a selector valve with three positions, as follows:

A. Neutral: flow in *E*, out *F*, the main piston is held in the center by two springs. Ports *J* and *K* are connected to the return connections *H* and *G*.

B. Cylinder up: the main piston moves to the right. Pressure in at *E* goes out through *J*, port *F* is shut off. Port *K* is open to *G* and *H*.

C. Cylinder down: the main piston moves to the left. Pressure comes in at *E* and goes out *K*. *F* is shut off. *J* is open to *G* and *H*.

In operating the valve, the piston is moved to the left by introducing pressure into chamber *N* and is moved to the right by introducing pressure into chamber *M*.

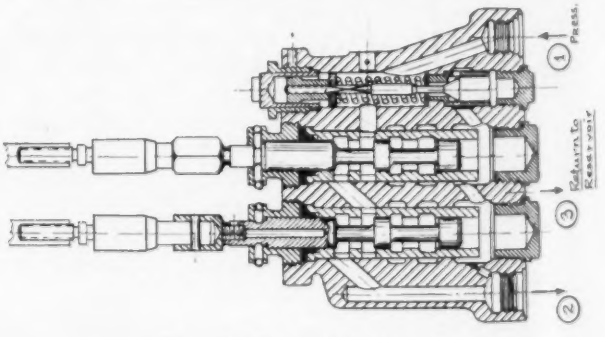
Pressure is introduced into chamber *N* by moving pilot piston *A* to the right. Pressure is introduced into chamber *M* by moving pilot piston *A* to the left.

Pilot piston *A* is spring centered in the neutral position when pressure is removed from both chambers *M* and *N*. Pilot piston *A* is moved to the right or left against its center springs by solenoids *P* and *Q*. Pilot piston *A* slides in inserted sleeve *B* which carries the porting.

The purpose of the pilot is to use small solenoids on the balanced pilot piston and let the pressure do the work of moving the large main piston.

Stacked type of slide selector valve: Fig. 14 shows a three-banked, stacked, slide selector valve with push-button operation. In it is incorporated a device for automatic return to neutral at the end of the actuating piston stroke. It is used for controlling the flaps, and incorporates an additional lever marked "start."

The automatic return to neutral device differs from that used by Elma on their push-button selector (already discussed), in that the Elma device depends on a buildup of pressure at the end of the piston stroke to kick the valve



■ Fig. 12 – Elma push-button valve

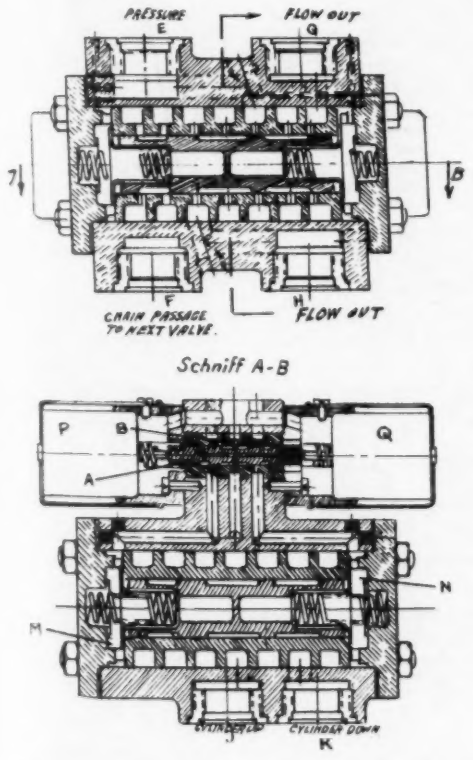
into neutral. The V.D.M. unit incorporates a differential diaphragm device which pulls out a detent pin (allowing the piston spring to return the valve to neutral) only after the fluid *flow* has stopped. This is an advantage over the Elma method, inasmuch as Elma had experienced considerable trouble with their valve returning automatically to neutral before the piston reached the end of its stroke, when extraordinarily high loads were encountered. Fig. 14 is a schematic cross-section indicating the method of operation.

The start button is incorporated so that the flap can be stopped at any midpoint of its travel. The construction and operation of the valve are apparent from Fig. 14. Some points to be noted are as follows:

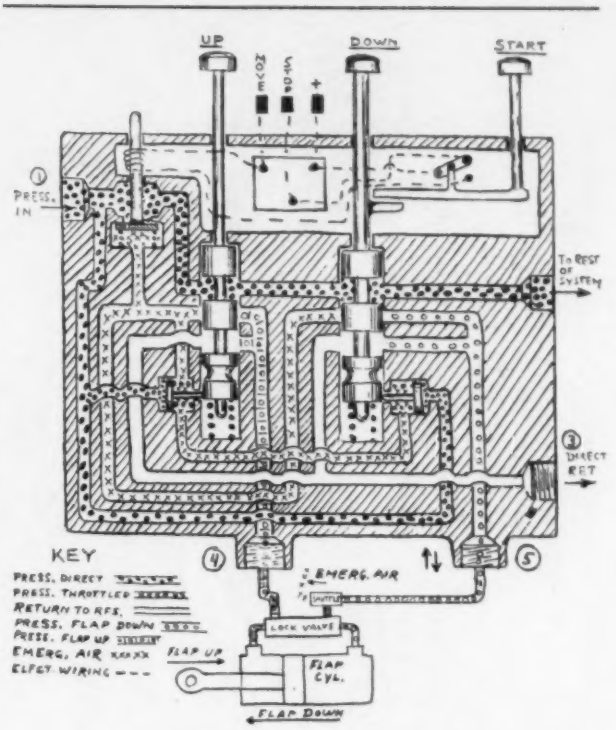
(a) In the neutral position of the valve the through ports (1-2, Fig. 14) are isolated from the cylinder ports (4 and 5). Also the two cylinder ports are both open directly to the reservoir (port 3).

(b) If, for instance, the "aus" button is depressed it will stay down (by the pressure differential detent device) until the piston reaches the end of its stroke. If, during the stroke, it is desired to return the flap to its up position, the "ein" button can be depressed, even though the "aus" button is still down. This action will immediately stop the flap cylinder and will cause the "aus" button to pop into neutral, the "ein" button remaining depressed until the flap is in its up position.

(c) If the start button is pushed, the piston will move only so long as the start button is held down, since in pushing the start button the solenoid is energized, which opens the throttle valve equalizing the pressure on the detent holding piston so that it will not hold the "aus" button depressed.



■ Fig. 13 – Teves solenoid pilot type of selector valve



■ Fig. 14 – V.D.M. selector (flaps)

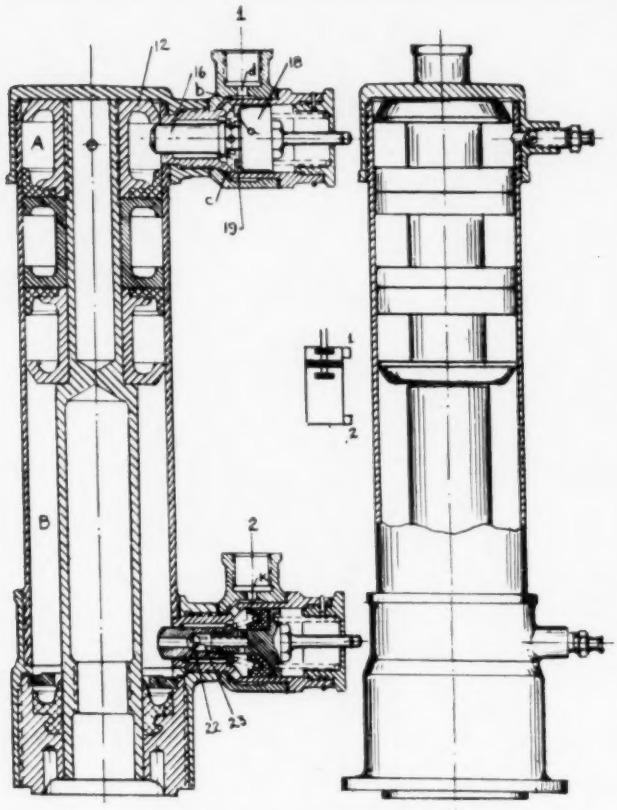
(d) Electrical contacts, shown in Fig. 14, to indicate the action of the cylinder.

Hydraulic Actuating Cylinders – Teves actuating cylinder: Actuating cylinders of the Teves design all follow the same general pattern. (See Fig. 15.) A special molded packing which acts as a U cup is used. The cylinder heads are screwed on using metal or fiber gasket seals. In permanent assemblies the heads are also screwed on, but in lieu of packing the screwed joint is brazed closed.

Where mechanical locks hydraulically released are needed, a different and more positive type of lock than either Elma's or V.D.M.'s is used. One of these piston locks is shown in the section on the right end of the cylinder in Fig. 15. When the piston reaches the end of its stroke, the locking piece 12 raises piston 16-18, allowing the piston to fall in behind the locking piece and lock the piston in position, as it is shown in Fig. 15.

In order to prevent any thermal expansion or other fluid pressure buildup from pushing the locking piston up and inadvertently unlocking the main piston, a thermal relief valve in the form of passage *K*, ball 22, and spring 23 is incorporated in the locking piston. The extension on the locking piston provides (if desired) a connection for a pull type of mechanical emergency unlocking means.

When hydraulic pressure is applied to port 1, it goes through bore *a* into chamber *b*, through bore *c* and into chamber *d* where it acts on piston 16-18 (16 and 18 are screwed together on assembly). (Spring 19 is only to retain packing.) This moves the locking piston up clear off the locking piece 12 and on up until oil flows out into chamber *A* around the locking piston. A breathing hole is provided in the chamber above the upper end of the locking piston to prevent any pressure buildup above the piston as a result of leakage past packing. Bleed connections are provided



■ Fig. 15 – Teves actuating cylinder

on many of the Teves cylinders. Fig. 15 shows a standard design with locking and bleed ports at both ends.

Actuating cylinder: Fig. 16 shows the landing gear cylinder on the Me 410. It is provided with electric switches (actuated by the piston) at either end to light signal lights.

On each end is also provided a sequence valve opened by the piston. On each end of the cylinder there is built into it a mechanical lock consisting of balls working against segments, which, going into grooves at the end of the cylinder actually lock the piston in place. Two spring-loaded pistons inside the main piston head hold the balls out against the locking segments, keeping the main piston locked until hydraulic pressure moves them back against their springs. This action unlocks the main piston. The locking segments, when they are in the locking groove, also operate the electrical switch for the signal lamps showing when the gear is actually locked up or down.

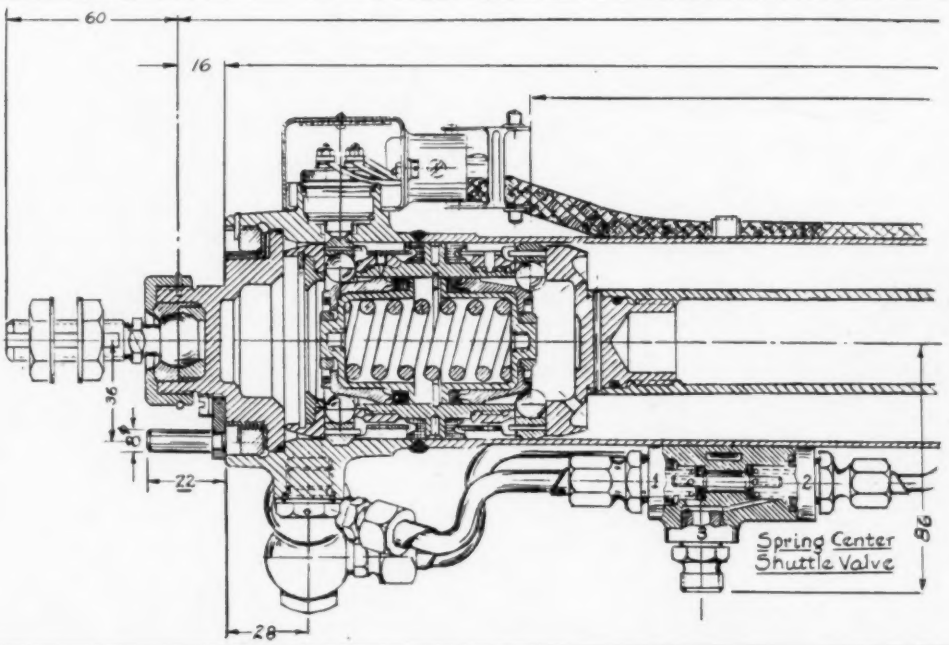
Attached to the cylinder, with ports 1 and 2 connected to the lower chamber of the two sequence valves is a spring-center shuttle valve. In the Me 410 circuit the outlet or port 3 of this shuttle valve leads to port 5 on the automatic flow-reversing sequence valve.

Actuating cylinder: Elma made actuating cylinders of all sizes and types, of which Fig. 17 is a good representative example. This hydraulically released mechanical lock is an Elma patent and was used extensively on German planes.

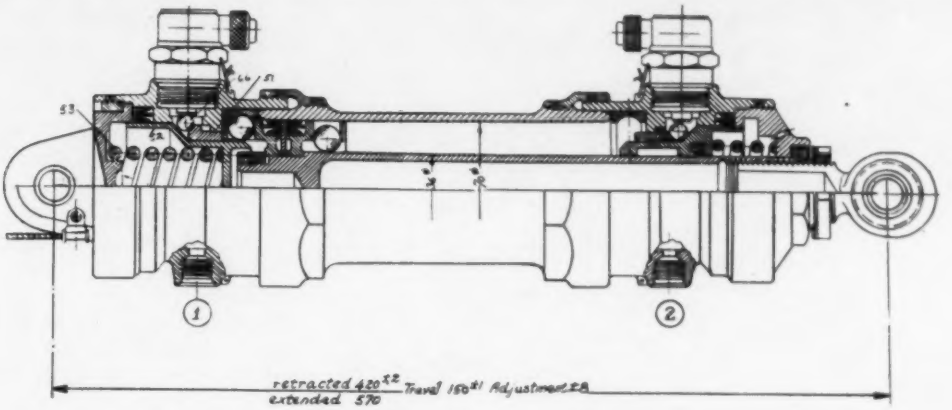
The electric limit switches on both ends of the cylinder actuated by the piston were also widely used. The switches themselves are of the snap action spring return to contact type. They are hydraulically sealed as far as the internals of the switch are concerned, so that they are merely screwed into the bosses on the ends of the cylinders with a straight thread gasket seal, such as any standard fitting. The small ball 66 is normally down in the hole clear of the switch. When the cylinder gets to the end of the stroke, it pushes against slide 51, which pushes small ball 66 up the taper, unlocking piston 52 against the switch, thus breaking the circuit, de-energizing the solenoid, and allowing the selector valve to return to neutral. This opens both ends of the cylinder to the reservoir.

As soon as pressure is off, spring 53 returns, locking piston 52 to position, sliding under the locking balls, and mechanically locking the piston. As soon as pressure is again applied (hydraulically), pressure will move locking piston 52 back first, thus unlocking the balls and permitting pressure to move the piston.

U cup packings are used throughout. The cylinder



■ Fig. 16 – V.D.M. actuating cylinder assembly



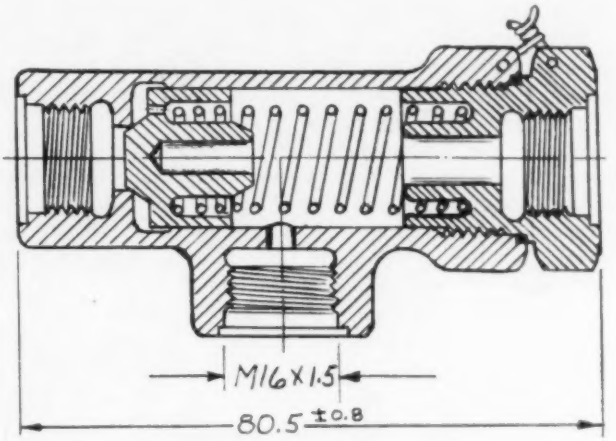
■ Fig. 17 – Elma actuating cylinder

heads are screwed on and locked by lock screws. One or two rings of balls can be used in the piston head, depending on whether a lock at only one end or both ends is required. The springs which actuate the locking piston are sealed off from hydraulic oil, and breather holes are provided in the spring chamber to let out any oil which might leak past the piston packings.

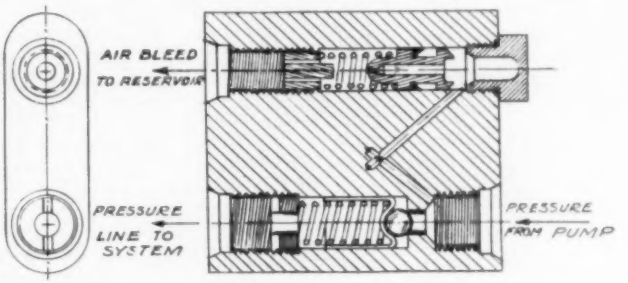
Attached also to the cylinder is an emergency valve, port 4 of which is connected with the lower chamber of the air sequence valve on the landing gear door cylinder. Port 1 of this emergency valve is connected to the upper chamber of the sequence valve on the landing gear downside of the landing gear cylinder. Port 2 on this emergency valve is connected directly to the side of the piston which will push the gear down. Port 3 is connected both to the gear-up port of the cylinder and to the upper chamber of the sequence valve on the up end of the landing gear cylinder. Port 4 is for emergency air.

Double U cup packings are used in both the main piston and the lock release pistons. They are also used in the piston rod packing. The cylinder heads are butt welded on to the barrel of the cylinder and are provided with removable plugs.

Special-Purpose Hydraulic Units – The automatic flow-reversing sequence valve (Fig. 4B) was used in the landing gear system of the Me 410. The operation of the landing gear circuit was as follows: After take-off the up button on the landing gear selector was depressed, allowing pressure to go to the door cylinder via the automatic sequence valve and the emergency valve and also directly to the stop chamber of one of the mechanical sequence valves on the door cylinder. Thus, the door cylinder would open the doors and, when the door cylinder piston reached the end of the stroke, it would open both mechanical sequence valves. During the operation of the door cylinder the return flow from it was going back to the selector valve via the automatic sequence valve. When the mechanical sequence valve on the door cylinder was opened by the piston, pressure went through this valve to the upper chamber of the mechanical sequence valve on the "gear up" end of the landing gear cylinder, thus raising the gear. During this process, the return flow from the landing gear cylinder goes via the emergency valve to the other sequence valve on the door cylinder (which is still in the open position), allowing this return flow to get to the reservoir by the same path described above for the return flow from the door cylinder.



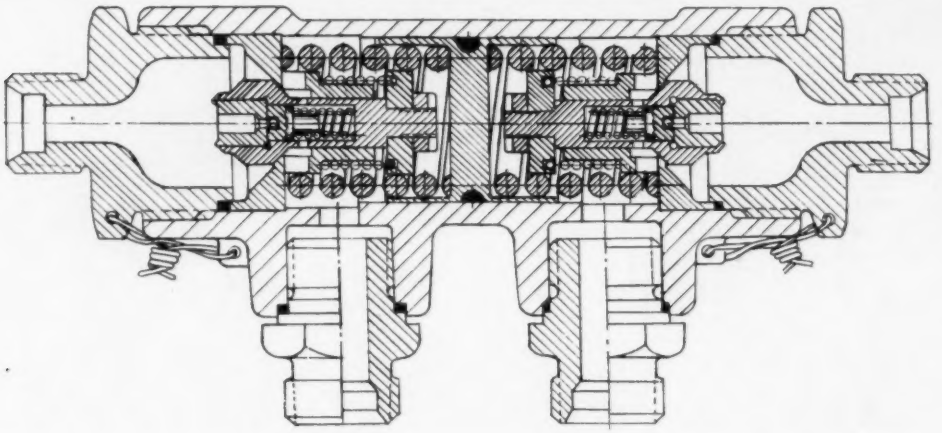
■ Fig. 18 – Elma simple shuttle valve



■ Fig. 19 – Elma air bleed valve for Do 335

When the gear is fully retracted the mechanical sequence valve on the "gear up" end of the landing gear cylinder is opened by the piston, allowing the pressure in the top chamber of this valve to go through port 5 to the reversing piston in the automatic sequence valve. This pressure acting against a 1200-psi spring in the automatic sequence valve piston reverses the flow to the door cylinder, thus closing the doors. When the door cylinder reaches the end of its stroke all flow ceases and this fact causes the selector valve to return to neutral, removing pressure from the entire circuit and allowing the 1200-psi spring to return the automatic sequence valve reversing piston to neutral.

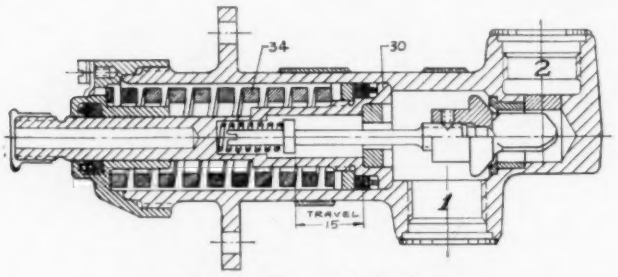
Spring-loaded shuttle valve: Fig. 18 shows a valve for use with emergency air systems. A spring holds the double-



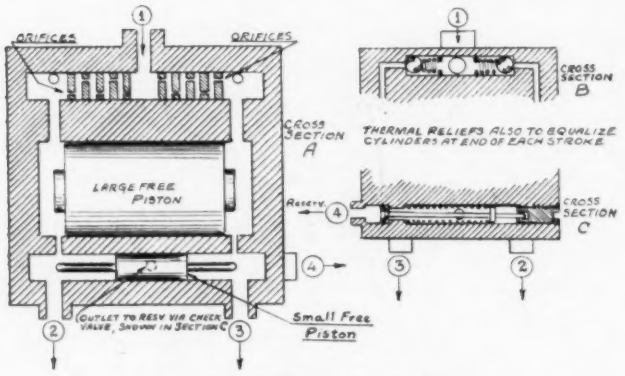
■ Fig. 20 – Elma double lock valve with integral pressure relief

seat poppet in its normal position permitting oil flow. When emergency operation is required, the emergency air pressure coming against the poppet pushes it off its normal seat. Since the poppet is a close fit, emergency pressure is not let out of the line to the cylinder until the poppet has cleared the cylinder port. Emergency pressure then goes out the cylinder port and also continues to force the poppet over until its other seat is reached and the normal port is sealed off. If the emergency pressure is released, or normal pressure restored, the spring will automatically return the poppet to its normal position.

Air bleed valve: Fig. 19 shows a valve used for the purpose of bleeding air from the hydraulic system in case some gets into the pump. It permits the pump to unload itself of air against no head, and thus prevents the pump from becoming air bound and burning itself out. During the normal operation, the top piston (see Fig. 19) is held to the left against the plug by oil pressure, thus closing the small bleed hole in the top piston. The check valve is open and oil flows through, as indicated in Fig. 19. If the pump gets air bound, naturally its pressure drops, causing the check valve to close, and since the pressure against the top piston is reduced, this piston under the action of its spring, moves to the right and opens the bleed hole, allowing air to bleed out to the reservoir through this piston. As soon as the pump has cleared itself of air, pressure again builds up, and pressure drop across the orifice or bleed hole in the top piston increases, forcing this piston again to the left against its spring, until it again contacts the plug which closes off the bleed hole. The check valve opens meanwhile and system resumes normal operation.



■ Fig. 21 – Elma control cylinder



■ Fig. 22 – Elma flow divider

Hydraulic lock-out valve, with thermal relief valves: Fig. 20 shows a conventional type of hydraulic lock-out valve for hydraulically locking an actuating cylinder at any point of the piston travel, when pressure is not actually being applied to either end of the cylinder (that is when the selector valve is in neutral and both cylinder ports are open, piston displacement in either direction will be acting against one or the other of the two check valves, thus holding them closed and keeping the piston locked). In the poppets of both of these check valves are incorporated small ball type of thermal relief valves. Between the inlet ports (from the selector valve) is a spring-centered packed piston of about four times the area of the check valve poppets. When the selector valve is put into the operating position, pressure coming to the lock-out valve opens one check valve directly, and by acting on the spring-centered piston moves it over against the other check valve, opening it and holding it open, thus allowing the piston to be moved hydraulically.

Control cylinder valve: This valve is used to allow free flow in one direction (port 2 to port 1) and to stop flow in the other direction (port 1 to port 2) until pressure available is equal to, or greater than, a predetermined amount. (In this case 20 atmospheres.) Furthermore, when the pressure has reached this predetermined amount the valve opens and allows full pressure, regardless of how high, to go out port 2. (See Fig. 21.)

It is often used where the load in a midposition may be high and where application of a lower pressure might allow reverse action. There are many other cases where

it is undesirable to have any pressure to a cylinder until the pressure available has reached a predetermined amount, and it is then desired to have the full pressure applied.

Another important application of this valve can be seen in the Me 262 hydraulic system, where it is used in series with the nose-wheel latch type of lock on the nose-wheel down end of the cylinder. In this installation, the extending piston rod is attached to, and acts as a release for the nose-wheel latch. When the pressure has reached a predetermined amount, it moves piston 30 back, releasing the nose-wheel latch. Only after this has been accomplished is the poppet lifted from its seat and the flow allowed to go to the nose-wheel-down end of the nose-wheel cylinder.

The detailed action of this valve is as follows: Pressure entering at port 1 acts on the poppet, keeping it closed. This pressure also acts on piston 30, tending to move it against heavy spring 34 away from the poppet. The poppet has an extension up into a hollow part of this piston so that when the piston has moved far enough it picks up the poppet extension and pulls the poppet off its seat against the pressure (since the piston area is considerably greater than the poppet area). Once the poppet is open, pressure gets under it and it is held open until the pressure drops considerably lower than the 20 atmospheres required to open it. In addition, the full available pressure gets into the cylinders, since there is no appreciable throttling of the flow, nor must the pressure work against the spring, as it would in an ordinary relief type of valve.

For return flow (port 2 to port 1) the pressure acts on the full seat area of the poppet, tending to open it against only the light spring by which the poppet extension is connected to the piston. During the return flow the piston, of course, is stationary.

Flow divider: This valve is used mostly in flap circuits. (See Fig. 22.) It is believed to be different in principle from any flow divider we have used. The principle is:

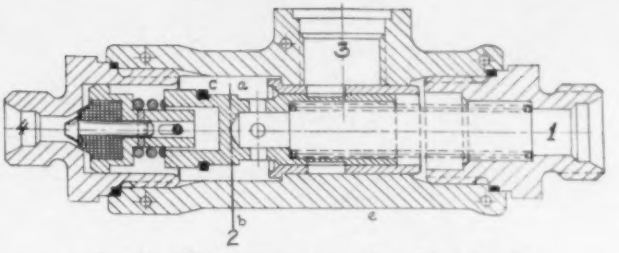
Pressure enters port 1, divides and goes through orifice plates into chambers at either end of the free piston (two thermal relief valves are connected from each outlet to port 1.) This free piston balances the pressure in the two chambers by moving across the outlet orifices to working cylinders. In this way it throttles the flow to the two cylinders. After the pressure gets past the orifices controlled by the free piston, it goes into the smaller chambers on either side of the lower small free piston. If the pressure is balanced, then the piston is centered and covers the small port, which leads via a check valve to port 4 and to the reservoir. If the pressure is not equal in these two chambers, the small piston moves left or right, opening the small port and bleeding pressure back to reservoir.

With a load differential of 12 to 1 on the two cylinders, Elma claims accuracy of 1.5% on flow requirements from 1 to 10 l per min.

Emergency valve with return flow: Fig. 23 shows an emergency valve used in the landing gear circuit of the Me 410. (See Fig. 4A for installation.)

Pressure enters at port 1 and goes out at port 2 or comes in at port 2 and goes out port 1 and does not affect the valve. Pressure at port 3 is blocked and does not affect the valve. Emergency air coming in at port 4 pushes back the piston, seats, sealing off the connection between ports 2 and 1 and going out port 2. At the same time, this pushing back of the piston connects port 3 with port 1.

An emergency valve such as this is required in the type



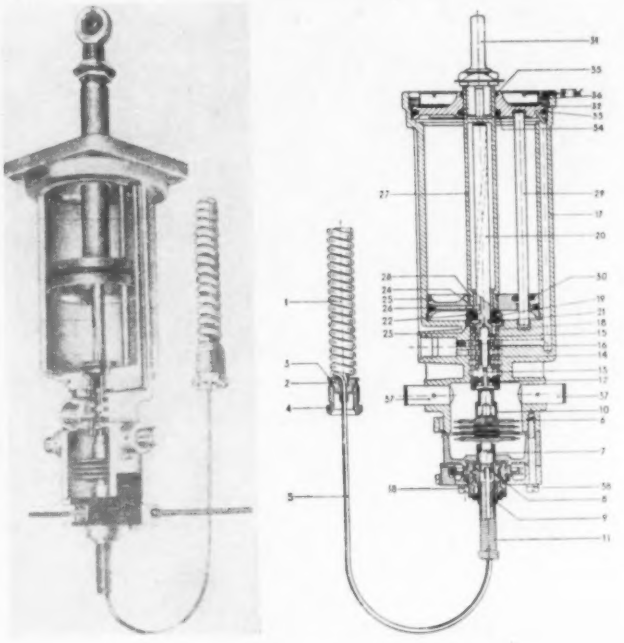
■ Fig. 23 – V.D.M. emergency valve with return flow

of circuit used in the Me 410 landing gear, since at any time during the complicated cycle it must be possible to open the landing gear doors and lower the landing gear by emergency air. To do this an emergency valve must, in addition to shutting off normal flow and introducing the emergency air to the proper side of the cylinder, open the return flow side of the cylinder directly to the return line, thus bypassing and eliminating the remaining part of the cycle. An inspection of Fig. 4A will make this clear.

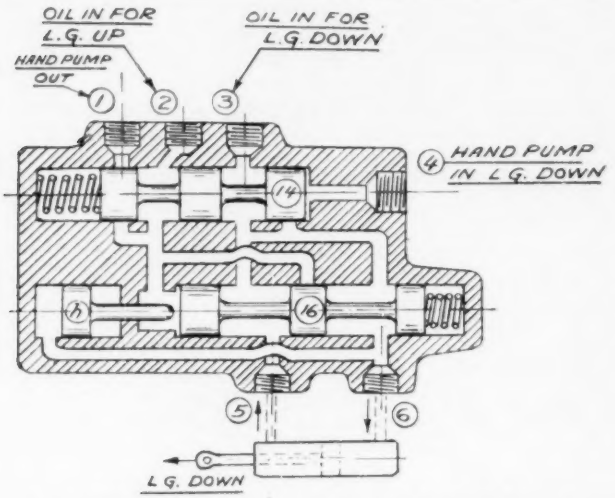
In this installation (for example, the door cylinder emergency valve) port 4 of the valve is connected directly to the emergency air line. (For the landing gear cylinder emergency valve port 4 is connected to the emergency air line via the mechanical air sequence valve of the door cylinder.) Ports 1 and 2 are for pressure in and out or for return flow in and out (depending on the position of the selector valve and the automatic flow-reversing valve). Port 3 is connected with the return line or pressure line (depending on the position of the selector valve and the automatic flow-reversing valve). Port 2 is always connected to side of cylinder to which it is desired to apply emergency pressure.

In the Me 410 landing gear circuit it is, therefore, connected to the open side of the door cylinder, and to the down side of the landing gear cylinder.

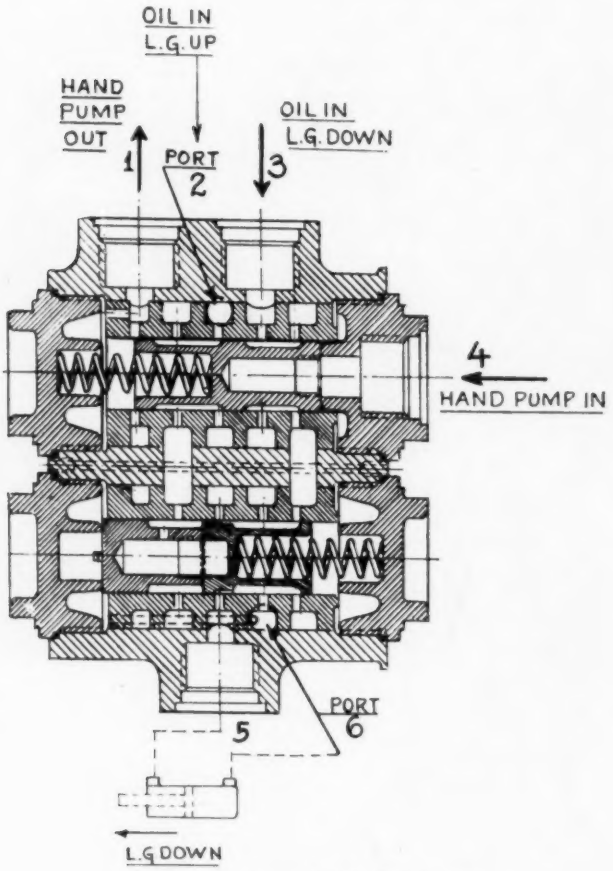
Thermostatic automatic cooling flap control cylinder:



■ Fig. 24 – V.D.M. thermocontrol cylinder



■ Fig. 25A – Schematic drawing of Teves double shuttle valve



■ Fig. 25B – Teves double shuttle valve

Fig. 24 shows a combination hydraulic cylinder with a small balanced servo and hydraulic follow-up device operated by a temperature bellows which used to control the engine temperature by automatically opening and closing the cooling shutters as needed.

Hydraulic pressure enters port 1, and port 2 is connected to the return or reservoir. With an increase in temperature above the predetermined temperature, the bellows expands, moves the small pressure balanced servo pistons upwards against its light spring. This bleeds pressure into the upper cylinder chamber and opens the low cylinder chamber to the return line, thus causing the piston to move downwards, opening the shutters.

The piston is kept from turning by a rod loosely held by the two ends of the cylinder. The piston rod itself is hollow, and another rod with two long spiral grooves fits up inside the piston rod. Two pins in the piston rod protrude inwards and engage these two spiral grooves. Thus, as the piston moves downwards, the spiral rod is rotated. The outer end of this spiral rod is threaded into the port sleeve in which the servo piston slides. The port sleeve is also kept from turning by a groove in it and a pin fixed in the body which rides in this groove.

Therefore, as the piston moves downwards opening the shutters, it turns in the threaded port sleeve, pulls this sleeve upward following the servo piston, and eventually bringing the valve into its neutral position, where the return line and the pressure line are shut off and the piston is locked by the liquid trapped on either side of main piston.

In the bellows end is a high pitch thread engaging an internal thread on a "telekin" control wheel. Since the bellows slide is constrained from turning, the position of the bellows, and hence the adjustment of the unit can be made manually. In the case of improper functioning, the pilot can manually override the unit by means of this "telekin" remote control system.

Ports 3 and 4 are provided in case the piston of the standard unit is not large enough to provide the necessary driving force. In this case a helping cylinder (nebenzylinder) is connected to those two ports. This connection puts this helping cylinder in parallel with the resulting cylinder, thus adding the necessary additional force needed but not affecting in any way the operation of the unit.

Automatic short-circuit valve: Fig. 25A is a schematic drawing in which the operation is more apparent than from the actual cross-section drawing (Fig. 25B).

Under normal operation piston 14 remains in the position shown. For landing gear down, pressure oil comes in port 3 and is admitted to both ends of the landing gear actuating cylinder through ports 5 and 6. If the weight of the landing gear and/or the slipstream effect is such as to tend to create a vacuum in port 4, the piston remains as it is shown in Fig. 25A, since no appreciable pressure is applied to the end of piston *H*. This allows flow to short circuit from one end of the cylinder to the other, so that only the oil needed to make up for the piston-rod displacement need be supplied by the pump. As soon as the gear reaches a point where pressure starts to build up in port 6, this pressure reacts against the end of piston *h*, which forces piston 16 to the right and thus closes the short-circuit path and opens port 5 to the return line port 2, and normal operation follows.

For landing gear up, pressure oil comes in port 2, immediately goes behind piston 16, moving it to right, opening pressure flow to port 5 and returning flow from port 6 to 3.

For emergency lowering of landing gear by hand pump, hand pump pressure enters port 4, moves piston 14 to the left, closing off connections to ports 2 and 3. It also opens the hand pump pressure to the valve. From here on it functions exactly as it did under normal operation.

Hydraulic Brake System Units – Accumulator, bag type,

brake: Fig. 26 shows a bag type of accumulator. It uses for its outer shell the same type of bottle in which oxygen is stored. The accumulator has incorporated in it an automatic bypass valve and filler combined. This automatically bypasses the accumulator if it should burst or be shot away by gunfire. It also keeps the accumulator in a bypass position until the bag has been fully inflated. In some of the newest models, the bypass valve was omitted because they felt it was not worth the added complexity.

Power brake control valve: Fig. 27 shows an excellent design for a power brake valve. It is nearly completely hydraulically balanced and is designed with large port areas and thus has a low pressure drop and minimum lag. The breathing or pressure regulating device needs only a very light spring, so that this spring has no appreciable reaction against the pedal. The valve has a large, builtin direct hydraulic loading cylinder for putting brake pressure directly against the pedal. This gives 80-85% hydraulic feel direct, with only 15-20% via the spring of the regulator. Only one sliding packing is used. This construction gives a minimum of friction and a minimum of hysteresis in the performance curve. The valve is small, light, compact, and simple to build.

Manual master brake cylinder: Fig. 28 shows also a simple, light, and cheap design. It has functioned very well in several aircraft. It contains its own reservoir and a unique valve for allowing filling and breathing of the reservoir and yet not allowing oil to squirt out of the breath hole. When it is desired to fill the reservoir, screw 45 is turned in, holding ball 5 on its seat. The reservoir is then filled. The filling automatically stops when the pressure in the reservoir reaches the pressure of the filling means, since a certain amount of air is trapped in the top of the reservoir. After the reservoir has been filled, screw 45 is turned back out and locked with lock nut 3. This allows the reservoir to breathe, but the fluid level in the reservoir has been held low enough so that there will be no squirting out of the fluid during breathing. This valve is interchangeable in the standard brake pedal with the power brake control valve already described.

Summary

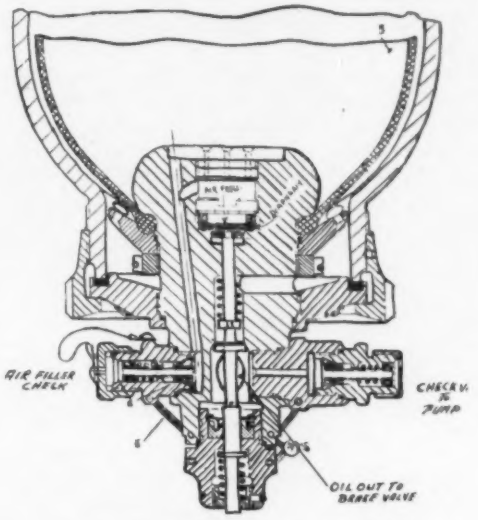
In conclusion it would be well to summarize briefly some of the most important German practices; especially those which might have a bearing on future hydraulics development in this country.

1. The German use of the open-center type of hydraulic systems eliminates the unloading valve and the accumulator so widely used in the United States. Their decision to use it was so well substantiated by test data and computations that serious consideration should certainly be given to its use in more of our aircraft designs.

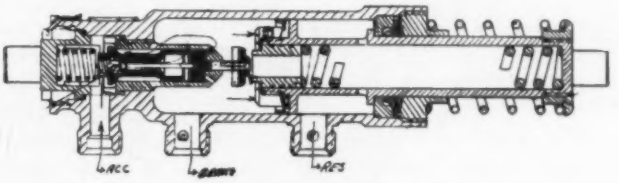
2. The simplicity in design of German hydraulic systems and particularly in the design of most of the components was so effective that our designers should at least give these items careful study and if possible actually try out some of the German ideas with samples and experimental systems.

3. The use of open-center systems with the resulting increase in permissible leakage of valves results in extensive use of the slide type of units, which simplifies the valves and reduces the cost in dollars and man-hours. It also permits complex operations and flow patterns to be used more frequently and safely.

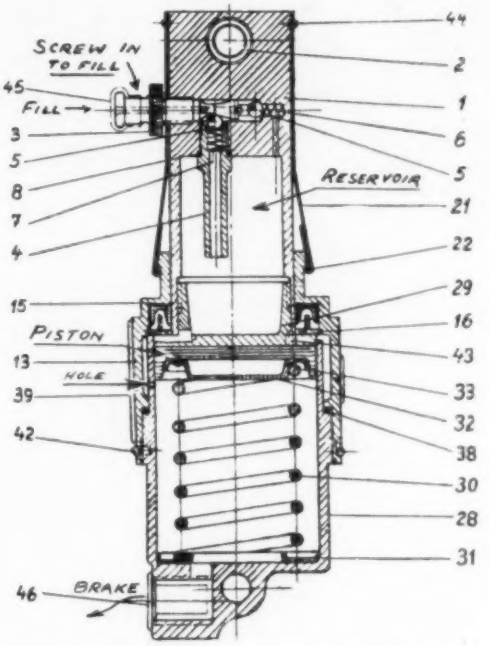
4. Although it is felt we were superior in many ways to the Germans in design of hydraulic components (especially from a mass production standpoint), in many of their accessories and in their general system design they showed enough ingenuity and originality, so that we cannot afford to overlook their work.



■ Fig. 26 – Elektron brake accumulator



■ Fig. 27 – Elektron power brake valve



■ Fig. 28 – Elektron master brake cylinder

SIGNIFICANCE of CETANE

CETANE number, a measure of the ingition quality of a fuel, is of importance in the satisfactory operation of a diesel engine; however, there is considerable evidence that cetane number is not the only criterion of a good diesel fuel and that other characteristics indicative of its burning properties must be considered.

The authors feel that basic research on diesel fuels is necessary to determine the influence of additives, hydrocarbon structure, and physical properties on ignition and combustion under a wide range of conditions. This work should be so fundamental, they believe, that the results would be applicable not only to diesel engines but also to gas turbines, jet propulsion engines, and rocket engines.

■

The opinions or assertions in this paper are the private ones of the authors and are not to be construed as official or reflecting the views of the Navy Department or the Naval Service at large.

THE problem of obtaining more power from a diesel engine of given weight and size has always been of great interest to the Navy. In the past, gains in engine output have been achieved by changes in engine design or materials, and the engines were required to operate on the fuels and lubricants that were then available. However, diesel-engine design has now progressed to the point where the use of superior diesel fuels may be justified in present Naval engines if certain military factors, such as reduction of smoke, increased reliability, and decreased maintenance, can be enhanced. Superior fuels will probably be a necessity for the high-output diesel engines of the future, just as high-octane gasolines are necessary for the satisfactory operation of the high-output gasoline engines in general use today.

■ Engine Design and Operation

When referring to high-output diesel fuels, the first characteristic usually mentioned is that of cetane number. The ASTM cetane number of a diesel fuel is defined by, and is numerically equal to, the percentage by volume of normal cetane in a mixture of cetane and alpha-methylnaphthalene that the fuel matches in ignition quality when compared by the procedure prescribed by ASTM designation D613-43T. Therefore, cetane number, by definition, is a measure of the ability of a fuel to ignite in the cylinder of an engine. This characteristic is of extreme importance in obtaining easy starting, smooth running at light loads, and in the reduction of exhaust smoke. Engine deposits, to some degree, are also a function of the cetane number of the fuels used, since fuels of poor ignition quality usually form a considerable amount of fuel residue. Thus, without question, the cetane number of the fuel is of definite significance in the satisfactory operation of a diesel engine. The cetane number of a diesel fuel can be compared with the volatility of a gasoline. A certain minimum cetane number is necessary in a diesel fuel, depending on the design of the engine in which it is used, just as a certain volatility is necessary in a gasoline to insure starting.

However, considerable evidence is accumulating which indicates that high cetane number is not the only criterion of quality in a diesel fuel, particularly from the standpoint of power output. Another characteristic, such as its rate of flame propagation or burnability in the cylinder, may be of equal or even greater importance for use in the high-output diesel engines of the future. While it is highly desirable that the diesel fuel ignite as soon as it is introduced into the cylinder, it is equally desirable that the fuel be controlled to burn as completely as possible during the early part of the power stroke, since combustion occurring during the final part of the stroke results in a loss of power as well as burned exhaust valves or ports, stuck piston rings, and high cylinder wear.

High cetane fuels produced by the Fischer-Tropsch process have been available for some time in Europe. Most of these fuels are essentially pure paraffins and are above 90 cetane number. Reports have been received that these fuels are not entirely suitable for use in present-day engines because (1) the rate of pressure rise is too low and (2) the rate of burning is too slow, resulting in after burning. These same fuels, however, when blended off with coal tar aromatics to give about 50 cetane number, are satisfactory for use in engines of present design.

A number of investigators in this country have also reported that the very high cetane fuels are not entirely satisfactory for use in engines of current design. Their data indicate there is no advantage in using fuels above approximately 60 cetane number and that the high cetane number fuels (above 90) are actually less suitable in present engines from the standpoint of power output and engine maintenance than fuels of 50 or 60 cetane number.

However, as far as is known, no data have been obtained in engines especially designed to take advantage of high cetane paraffinic fuels or of high-cetane doped fuels containing combustion accelerators.

Preliminary work on these high-output fuels seems to indicate that engine modifications will be necessary to take full advantage of the fuel. These modifications probably will consist of a reduction in compression ratio by an amount dictated by the ability of the engine to start and then supercharging to obtain the maximum pressures desired. It is possible that the turbocharger will prove to be the most suitable type of supercharger for this purpose.

Engine data obtained so far have indicated that a 20%

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NUMBER in FUELS

by COM. E. F. GRIEP, USNR*
and LT.-COM. C. S. GODDIN, USNR
Bureau of Ships
Navy Department

THE AUTHORS: E. F. GRIEP (M '38) has recently rejoined the Standard Oil Co. of Calif. Until recently a commander with the USNR, he served with the Internal Combustion Engine Section, Bureau of Ships in Washington. He joined the Standard Oil organization in 1929 shortly after his graduation from the University of Washington and was engaged in research and development until he left to enter the Navy in 1942. LT.-COM. C. S. GODDIN, USNR, is at present serving overseas on a special mission. He was until recently stationed with the Special Fuels Section of the Bureau of Ships. Before being commissioned in the USNR in 1941, Commander Goddin was for four years a chemical engineer in the Research Department of the Standard Oil Co. of Ind. He was graduated from the University of Michigan in 1937.

increase in output can be secured by decreasing the compression ratio to about 12-1 and supercharging to bring the maximum cylinder pressures up to that obtained at existing diesel-engine compression ratios. A fuel with a minimum of about 75 cetane number is necessary to insure easy starting under these operating conditions. These data also show that there is a considerable variation in the power that can be obtained from an engine while using two different fuels of the same cetane number. This variation is attributed to differences in the chemical composition of the fuels which affect their rate of burning.

Fuel Composition

The high-speed diesel engine requires a fuel which can be ignited and smoothly burned in a very short period of time. For this reason a study of both ignition delay and postignition burning characteristics is of importance in diesel fuels. From the fundamental standpoint, both ignition delay and burning characteristics are dependent on the following principal independent variables:

1. Molecular structure and physical properties of fuel.
2. Fuel additives, or internal ignition or combustion catalysts.
3. External ignition or combustion catalysts.
4. Composition of reaction mixture (fuel/air ratio).
5. Temperature.
6. Pressure.
7. Mechanical effects (such as combustion-chamber design, turbulence, spray pattern, and timing).

Items 1 and 2 are characteristic of the fuel alone and items 3-7, inclusive, are controlled by the engine or combustion-chamber design and operation.

To date, most attention has been centered on the study of the ignition quality of diesel fuels, especially the cetane number. Little data are available correlating burning characteristics with fuel composition. For a given type of fuel there may be a relationship between cetane number and burning characteristics, but it has been shown that this relationship can be altered by changes in fuel composition.

In addition to the cetane rating, other laboratory methods for studying ignition characteristics have been devised, particularly in constant-volume bombs.[1] Such methods are of great value in studying ignition delay and combustion phenomena under carefully controlled conditions.

A limited amount of data has been published on investigation of ignition quality as related to hydrocarbon structure. Some of these results are summarized in Tables 1 and 2.

The data in Table 1 were recently reported by Petrov and associates on the basis of a systematic study of the dependence of cetene numbers (as measured in the M.B. Nieman bomb) and pour points on the structure of hydrocarbons in the diesel fuel range.[2, 3] This work is apparently connected with development of a fuel with low pour point suitable for use in high-speed aviation diesel engines. It is noted from the data that transition from straight-chain to branched-chain paraffins reduces the pour point, but also reduces the cetene value. However, the cetene number is increased by increasing the length of

Table 1 – Cetene Numbers of Pure Hydrocarbons
(Data of Petrov and associates)

Name	Formula	Structure	Pour Point, F	Cetane No.*
n – dodecane	$C_{12}H_{26}$	C_{12}	−10	82
3 – ethyl decane	$C_{12}H_{26}$	$C_2-C(C_7)-C_7$, C_2	−94	55
2, 5 – dimethyl undecane	$C_{13}H_{28}$	$C-C-C-C(C)-C-C_6$, C	−94	66
4 – propyl decane	$C_{13}H_{28}$	$C_3-C(C_6)-C_6$, C_3	Below −85	45
5 – butyl nonane	$C_{13}H_{28}$	$C_4-C(C_4)-C_4$, C_4	Below −94	61
7 – butyl tridecane	$C_{17}H_{36}$	$C_6-C(C_6)-C_6$, C_4	−94	80
7 – hexyl pentadecane	$C_{21}H_{44}$	$C_6-C(C_6)-C_6$, C_6	−78	95-97
9 – heptyl heptadecane	$C_{24}H_{50}$	$C_8-C(C_8)-C_8$, C_7	−17	100

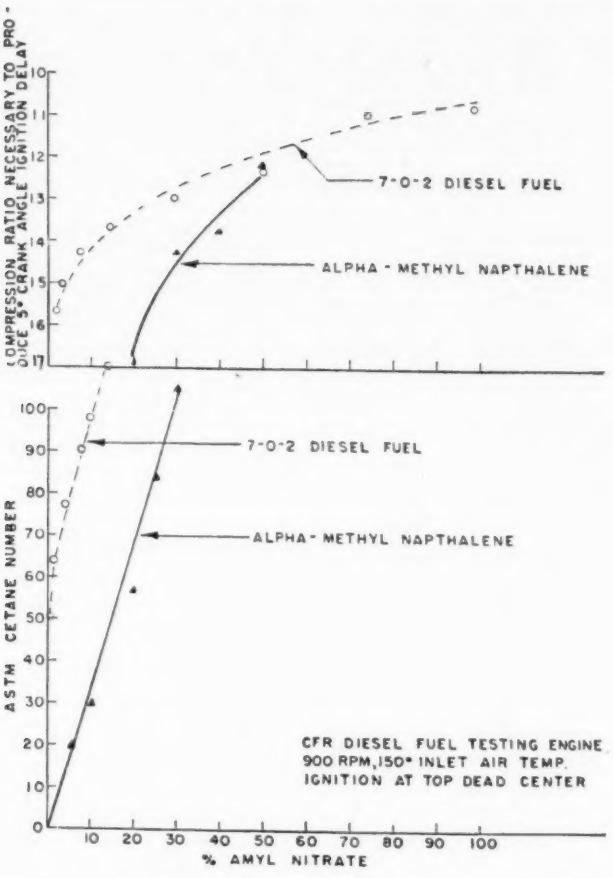
* Determined in the M. B. Nieman bomb.

* Now with Standard Oil Co. of Calif.

[1] See NACA Technical Report No. 617 (1938), "Autoignition and Combustion of Diesel Fuel in Constant-Volume Bomb," by R. F. Selden.

[2] See *Zhurnal Obschei Khimii*, Vol. 11, 1941, pp. 1104-1106: "Synthesis of 3-Ethyldecane and 2,5-Dimethyl Undecane," by A. D. Petrov, A. M. Pavlov, and Y. A. Makarov (U.O.P. Survey of Foreign Petroleum Literature Translation No. 451, July, 1944).

[3] See Izvestiya Akademii Nauk USSR, Otdelenie Khimicheskikh Nauk, 1944, pp. 238-242: "Synthesis and Properties of Isoparaffin Hydrocarbons with from 13 to 24 Carbon Atoms in Molecule," by A. D. Petrov and M. V. Vittikh (U.O.P. Survey of Foreign Petroleum Literature Translation No. 491, February, 1945).


COMPRESSION RATIO NECESSARY TO PRODUCE 5° CRANK ANGLE IGNITION DELAY
7-0-2 DIESEL FUEL
ALPHA-METHYL NAPTHALENE
ASTM CETANE NUMBER
7-0-2 DIESEL FUEL
ALPHA-METHYL NAPTHALENE
CFR DIESEL FUEL TESTING ENGINE
900 RPM, 150° INLET AIR TEMP.
IGNITION AT TOP DEAD CENTER
% AMYL NITRATE

■ Fig. 1 – Effect of amyl nitrate on ignition quality of diesel fuels

chains. It was concluded that molecules with a single long side chain in the center were of most interest as low pour point constituents of diesel fuel. It is worthy of note that the Russians considered fundamental work of this nature of sufficient importance to warrant its continued prosecution during the war under the auspices of the U.S.S.R. Academy of Science.

In Table 2 are summarized additional data on pure hydrocarbons obtained at the University of Birmingham.[4]

In comparing n-cetene with n-cetane it will be noted that the presence of the double bond results in a poorer ignition quality. Aromatics like benzene and toluene have poor ignition quality, but by adding a paraffinic side chain, ignition quality intermediate between paraffinic and aromatic is obtained, and the cetane number can be progressively increased. An interesting variation is shown in the comparison of biphenyl and its homologues, diphenyl methane and dibenzyl. Here it is noted that the more aromatic biphenyl possesses a higher cetane number. Addition of hydrogen to biphenyl to give the naphthenic bicyclohexyl markedly improves the cetane rating. The naphthenes deserve further study as a source of relatively high cetane – low pour diesel components. Another interesting comparison is that of n-hexyl benzene and di-iso-propyl benzene which have the same number of carbon and hydrogen atoms, yet the latter has an appreciably lower cetane number due to the greater number and smaller length of chains attached to the aromatic ring nucleus.

Attempting to generalize on the basis of the limited data presented, the following tentative conclusions for hydrocarbons are drawn:

1. Straight-chain paraffins have the best cetane numbers, and the longer the chain the higher the cetane, up to a certain maximum.
2. Aromatic rings will have the lowest cetane number.
3. In the case of branched paraffins or aromatics it can be said that the lower the number of chains, down to a minimum of one, and the greater the length of these chains, the higher will be the cetane number.
4. The greater the number of hydrogen atoms per molecule the higher the cetane number, although structural arrangements may overrule this: thus, naphthenes will be better than aromatics or cyclic olefins, and paraffins better than the corresponding aliphatic olefins.

An alternate method for improving the ignition quality of fuels is the use of special additives. Considerable work has been done on this subject and an excellent survey of the literature and summary of the status of this work is given in a recent article by Bogen and Wilson.[5] Most of the additives studied have been relatively unstable nitrogen or oxygen compounds, which apparently decompose under engine conditions to accelerate the fuel ignition reactions.

Fig. 1 summarizes preliminary data obtained at the U. S. Naval Engineering Experiment Station on the effect of additions of commercial amyl nitrate to alpha-methyl naphthalene and to a straight-run 50-cetane diesel fuel from natural petroleum. Since values of cetane number above 100 are meaningless, the ignition quality for blends above 100 cetane number are expressed in terms of compression ratio required to give a 5-deg crank angle ignition delay on the CFR engine. It is seen that the addition of about 11% of amyl nitrate is required to raise natural diesel fuel to 100 cetane number and about 30% to raise the alpha-methyl naphthalene to 100 cetane number.

It is obvious that a new yardstick for measuring ignition quality of fuels above 100 cetane is desirable. One suggestion is that they be rated in accordance with the percentage of amyl nitrate in cetane which exactly matches them in ignition quality.

Considerable opportunity exists for improvement of additives, particularly in the direction of improving storage stability and low-temperature starting characteristics. The effect of additives during the combustion period is not well established. In the case of amyl nitrate, one source reports a negligible effect on duration of combustion. Other data show, by the use of ethyl nitrate, an increase in the primary burning rate, but little effect on the secondary rate.[5] In any event, as mentioned above, the use of additives offers definite promise for control of combustion by permitting the use of lower cetane base fuels with advantageous burning characteristics.

■ Combustion Research Needed

In the case of the gasoline engine, improvement in performance due to design changes has been achieved coincident with development of superior fuels. The modern aviation engine utilizes a fuel of carefully controlled com-

[4] See *Oil and Gas Journal*, Vol. 39, Jan. 9, 1941, pp. 41 +: "Cetane Number and Chemical Composition," by T. Y. Ju and C. E. Wood.

[5] See *Petroleum Refiner*, Vol. 23, July, 1944, pp. 118 +: "Developments to Raise Ignition Qualities of High-Speed Diesel Fuels Explained," by J. S. Bogen and G. C. Wilson.

position, the major components of which are pure synthetics. Preceding and orienting the progress of gasoline development there has been a large amount of research on combustion characteristics of pure hydrocarbons in the gasoline range.[6]

On the basis of this work, great advances have been made in fuel technology, and the petroleum refining industry has made enormous investments in special equipment for the prime purpose of improving the quality of gasolines.

In contrast, on the other hand, is the diesel fuel picture. Comparatively little research has been done with a view towards developing superior diesel fuels. Especially with regard to basic combustion studies of hydrocarbons in the diesel range, research in this country is at about the level of gasoline research in the 1920's.

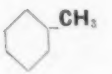
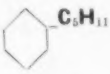
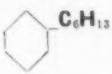
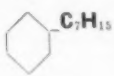
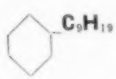
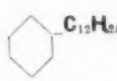
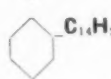
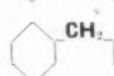
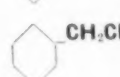
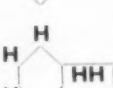
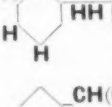
The development of the high-speed diesel has stimulated interest in the possibility of securing still better performance by use of tailored fuels. Interest in fuel composition is being further intensified by the fact that the demand for gasoline and the steady development of processes for converting gas oils into gasoline, especially wide-scale installation of catalytic cracking during the war, is tending to divert conventional straight-run diesel oils into gasoline production and to substitute therefor cracked diesel oils of altered composition and combustion characteristics.

On the basis of the preceding discussion it is considered that an accelerated program of research on diesel fuels is required to determine the influence of additives, hydrocarbon structure, and physical properties on ignition and combustion under a wide range of conditions. The variables measured should be fundamental enough so that the results would be applicable to the design not only of diesel engines, but also of other liquid fuel burning mechanisms, such as the gas turbine, jet propulsion engine, and rocket engine.

It is considered that emphasis should be put now on the study of pure hydrocarbons and additives. Work of this nature, which is already started on ignition quality, should be extended to quantitative measurement of time, temperature, pressure, and composition relationships during the whole process of combustion, including the periods both preceding and following ignition. Further development of basic physical-chemical theory concerning the combustion phenomena would be a logical corollary of this work. The job is important enough to warrant cooperative effort between the research laboratories of engine manufacturers, refiners, universities, and the Government. The results of this combustion research would be of tremendous value and stimulus to the engine manufacturer in the design of superior engines and to the refiner in the production of superior fuels.

One of the handicaps thus far in research on diesel fuels of petroleum origin is the difficulty of analyzing the complex mixtures of hydrocarbons in the diesel boiling range. Possibly the most suitable method available to date is that of Deanesly and Carlton whereby fuel composition can be determined in terms of per cent of carbon atoms in naphthene rings, aromatic rings, and alkyl chains.[7] Further research work is required, however, to develop more rapid and accurate means of typing diesel fuels. In the case of diesel oils of fairly simple composition, such as the Fischer-Tropsch oil, work at the U. S. Naval Research Laboratory has shown ultraviolet and infrared spectroscopy to be very informative.

Table 2 – Cetane Numbers of Pure Hydrocarbons (Data of Ju and Wood)

Name	Formula	Structure	Cetane Number: Pure	Cetane Number: Blending[a]
n-heptane	C_7H_{16}	C_7	57	
n-cetene	$C_{16}H_{32}$	$C{=}C\text{-}C_{14}$	88	
n-cetane	$C_{16}H_{34}$	C_{16}	100	
benzene	C_6H_6			−10
toluene	C_7H_8			−21
n-amyl benzene	$C_{11}H_{16}$		8[b]	18
n-hexyl benzene	$C_{12}H_{18}$		26	32
n-heptyl benzene	$C_{13}H_{20}$		35	39
n-nonyl benzene	$C_{15}H_{24}$		50	51
n-dodecyl benzene	$C_{18}H_{30}$		68	68
n-tetradecyl benzene	$C_{20}H_{34}$		72	72
biphenyl	$C_{12}H_{10}$			21[b]
diphenyl methane	$C_{13}H_{12}$			11
dibenzyl	$C_{14}H_{14}$			1[b]
bicyclo hexyl	$C_{12}H_{22}$		53	53
m-diisopropyl benzene	$C_{12}H_{18}$		−12[b]	−3

[a] In 50/50 blends. [b] Extrapolated values.

■ Production of Special Diesel Fuels

The modern refining industry has a great flexibility and

continued on page 448

[6] See *Industrial and Engineering Chemistry*, Vol. 36, December, 1944, pp. 1079-1085: "Effect of Molecular Structure of Fuels on Power and Efficiency of Internal-Combustion Engines," by C. F. Kettering.

[7] See *Industrial and Engineering Chemistry* (Analytical Edition), Vol. 14, March 15, 1942, pp. 220-226: "Type Analysis of Hydrocarbon Oils," by R. M. Deanesly and L. T. Carlton.

AUTOMATIC and SEMIAUTOMATIC

by HARRY R. GREENLEE

L. G. S. Spring Clutch Corp.

THE author proposes to bring the reader by steps into the realization that an automatic or power shifted transmission can be designed with the aid of today's manufacturing techniques and with fundamentals most generally known and in use. While the type of transmission generally discussed in this paper is for pleasure car use, it should be borne in mind that for a truck or bus it actually becomes a necessity, from the standpoint of shifting without breaking the drive line and without overlap. What could be more desirable for the truck operator than to be able to drop to lower ratios or raise to higher ratios progressively without engine raceaway or perceptible bump?

The step-type transmission, as compared with infinitely variable ratio types, may be more universally accepted in that today's drivers are accustomed to the step feel, leaving it up to the engine to produce the variable torque and speed during each shift. The author believes that, given a chance to choose between infinitely variable torque and a step-type transmission, the majority of pleasure car drivers would choose the latter; the importance of the feel of getaway to the driver being not unlike his sense of steering.

■ Automatic Transmissions

It is a temptation to make general statements in writing or in conversation that are attractive to the casual reader or listener, but the precision and accuracy of engineering data make for bulk, as well as for dull reading to most people, including many engineers; therefore, the text following is a pointed review of the basic fundamentals which the author believes to be the prime factors governing the construction of the most economical, simply constructed, easy to service, full or semiautomatic, power shifted transmission.

Much has been said and not a few attempts have been made to obtain the theoretically perfect transmission, one which produces infinitely variable torque and which may be tailored to the engine. True, it is the engineer's desire to reach infinite ratios and optimum performance, but, if the result is too expensive and the overall efficiency is inherently sacrificed, then what is to be gained by such a departure from one which may be more conventional? There will be time enough after the successful compromise is reached to let evolution carry the project through to an ever-improved transmission. A good engineer will always try to beat his own design.

[1] Overlap occurs when shifting from one gear ratio to another by holding one set of gears from disengagement until the other set is ready to pick up the load completely. Too much overlap results in a bump, while too little results in engine race-away, the latter being classed as a break in the drive line.

[This paper was presented at a meeting of the Indiana Section of the SAE, Indianapolis, Ind., Dec. 13, 1945.]

As a rule there is nothing new under the sun when it comes to producing new and novel combinations of old ideas. We think of radar and atomic power as something entirely beyond our ability to understand, because most of us are wholly unfamiliar with it. With automatic transmissions it may seem just as strange or new to many who are well versed in every individual mechanical feature in the design, simply because the puzzling combination of proved ideas is so broad that not even the best of mechanical minds can quickly grasp its overall importance.

■ Transmissions Now Vital Subject

This is one of the broadest subjects and most discussed topics that has struck the automotive field in many years. Its romantic appeal, accentuated by war developments, has brought to light every conceivable type of power transmission in order to fulfil war emergencies, engineering desires, new sales appeal, and, to give the automatic-minded driver a mental, as well as physical, ease of gear shifting never before experienced.

Automatic transmissions, without a doubt, will be one of the most outstanding developments in the immediate postwar period.

■ Transmission Classification

Transmissions may be divided into several classes, and combinations of these classes can be varied to suit individual tastes as well as a proper selection for the problems involved. More often than not the combination is a compromise between weight, overall dimensions, competitive sales requirements, performance, and price.

Transmissions basically can be classed in one or more of the following groups:

1. Standard 3- or 4-speed hand-shifted type.
2. Semiautomatic, where one or more of the upshift speeds is vacuum or solenoid selected by releasing the accelerator long enough for speeds to cross.
3. Full automatic *with overlap*. Usually 4-speed type, no clutch pedal, will power upshift or power downshift under full torque load, and is somewhat torque responsive. Clutch wear in some designs must be corrected by periodic retiming in the field by *factory trained men*.
4. Full automatic *without overlap* and *without breaking* the drive line.[1] May be three or four speeds, a minimum

TRANSMISSIONS

AN automatic or power shifted transmission can be designed, the author believes, with the aid of available manufacturing techniques and a fundamentally sound arrangement of known mechanical combinations.

Such a transmission, the author says, will upshift or downshift while under full torque load and without necessity of overlapping clutches and without breaking the drive line.

Mr. Greenlee considers such a transmission essential on trucks and heavy-duty equipment where shifting with heavy loads at low momentum interferes with present types of such transportation. He also sees a marked trend toward this type of transmission as a basic passenger-car requirement.

THE AUTHOR: HARRY R. GREENLEE (M '41) has been engaged during his professional career with the design and development of almost every type of passenger-car automatic transmission. With the L.G.S. Spring Clutch Corp. for more than two years, he has been successively development engineer, assistant chief engineer and has recently been appointed chief engineer. Before joining L.G.S., he spent four and one-half years with the Studebaker Corp., in the development of automatic transmissions, drive lines and overall vehicle construction. For some time prior to that, he was associated with the Detroit Gear Division of the Borg-Warner Corp.

of power loss when power shift is made, and no clutch wearout timing adjustment is ever necessary. Somewhat torque responsive and little field service required.

It is needless to add that many variations branch out from the above types, such as the use in combination of: fluid flywheels, torque converters, vacuum actuated clutches, hydraulic controls, electric governors, solenoid controls, and brain boxes.

■ Today's Semiautomatic Transmission

Fig. 1 shows the evolution of a conventional 3-speed, countershaft type of transmission in which a fluid coupling has been added to the front end, while the semiautomatic planetary overdrive has been attached to the rear end.

This type of transmission lends itself to many types of semiautomatic controls and attachments. However, one can note the apparent "add a unit" features as compared with one that would be purposely designed in a single unit to produce the same or better results.

■ Today's Full-Automatic Transmission

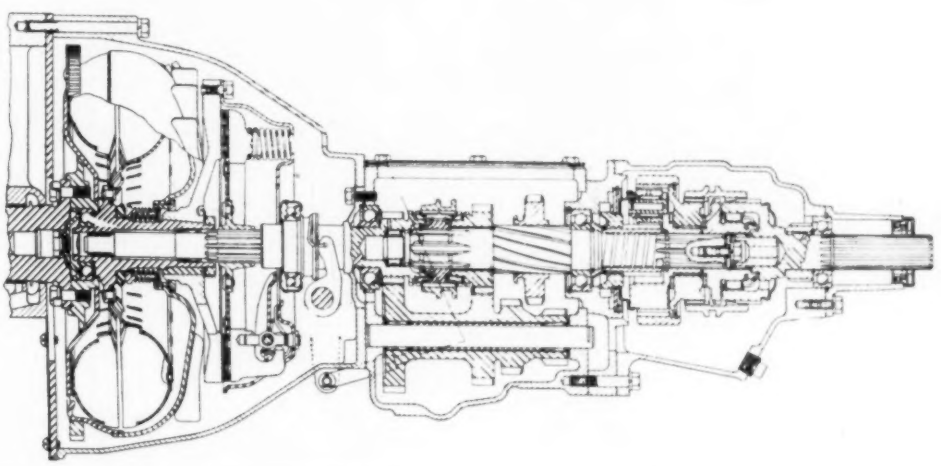
Fig. 2 shows a 4-speed planetary transmission using multiple-disc clutches in oil, together with external bands, all being hydraulically and automatically controlled. A fluid flywheel is used. Obviously, it is a more compact design in that it was built with automatic features in mind in the first place. This transmission will shift under full torque load but with controlled overlap during the shifts.

■ Semiautomatic Transmission Construction

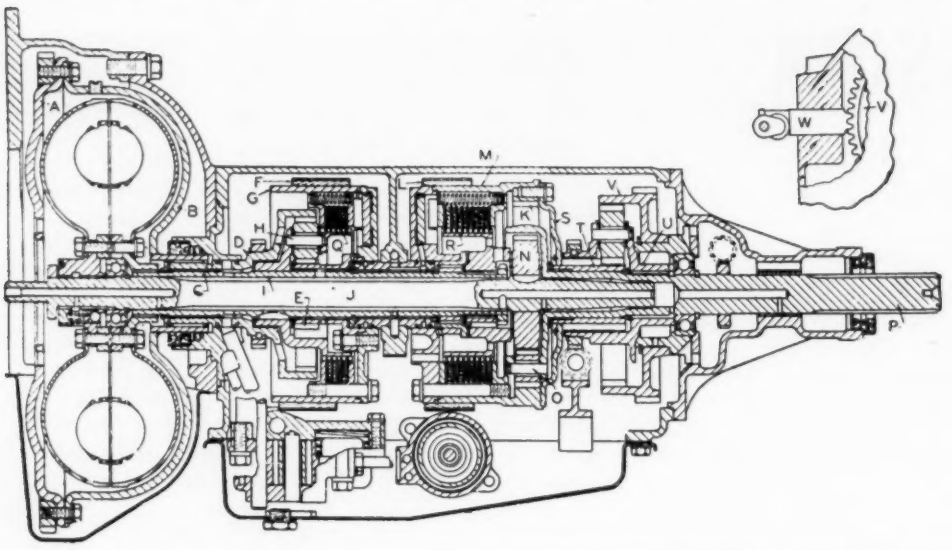
Today's conventional 3-speed transmission with semiautomatic overdrive has been very widely accepted and has proved practical for nearly every type of pleasure car driving. (Same as Fig. 1 without fluid coupling.)

For level road driving one can start the car in second gear, release the accelerator above a predetermined speed and, as the shaft speeds cross, the transmission will be in overdrive second, which is a little higher torque ratio than direct or third speed. When the desired pickup is reached the shift lever is placed in high gear position, with the aid of the master clutch, resulting in an overdrive ratio of 0.7222 revolutions of the engine to 1 of the propeller shaft.

At this time (or even while in second overdrive) the operator may depress the accelerator to the floor to effect



■ Fig. 1—Three-speed transmission with overdrive



■ Fig. 2 – Automatic 4-speed planetary transmission

a downshift. In the process the ignition is interrupted two or three explosions, sufficient to create a momentary torque reversal, which permits the sun gear to be unlocked, thus dropping the load to the oneway clutch; and the planetary overdrive will be in direct drive or one speed ratio lower than before.

At first glance one would ask why not two such units combined, resulting in a 4-speed, semiautomatic transmission. It is true that we have a semiautomatic transmission and variations have, from time to time, been attempted, but so many unsatisfactory results have developed from a cost and performance standpoint that they were discarded because the bad features outweighed the good.

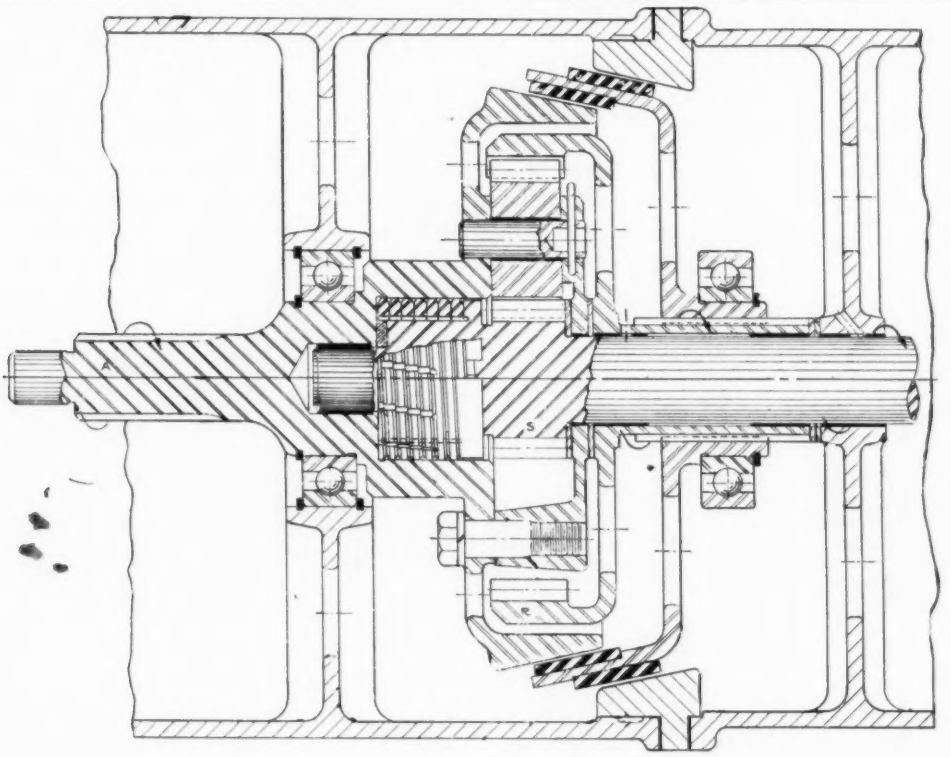
We have, however, learned the following from the above venture:

(a) Planetary gearing lends itself to the flexibility of automatic operation.

(b) A step increase in gear ratio in the order of 35% can be made without perceptible jerk or bump to the passengers when downshifted.

(c) To wait for speed crossing at lower ratios where shifting time should be at a minimum would take a longer period of time than one would be willing to wait.

(d) We have a shift *without overlap*, but there is the equivalent of a break in the drive line in that the engine must be decelerated during upshift or downshift.



■ Fig. 3 – Overdrive

Naturally, we will conclude that a shift without breaking the drive line and without interrupting the torque output in the rear wheels would be a basis from which to start a transmission of the type described in classification No. 4.

■ DeNormanville Overdrive Or Underdrive

In 1940 Capt. DeNormanville demonstrated to many automotive engineers his overdrive transmission which could be shifted under full torque load without overlapping clutches and without breaking the drive line. A brief description follows using the overdrive design. (See Fig. 3.)

This is a simple planetary gear set in which the driving shaft *A* is the planet pinion carrier, the ring gear *R* is the reaction member, and *S* is the sun gear or output shaft.

Interposed between the input and output elements is a oneway spring clutch, while a double-acting cone clutch is adapted to clutch the reaction member *R* (ring gear) to the case for ratio or to couple the input and reaction numbers together for direct drive.

The ratio may be 0.722 revolutions of the input shaft to 1 revolution of the output shaft or its reciprocal of 1 revolution of the input shaft to 1.38 revolutions of the output shaft.

When the ring gear clutches against the case to prevent rotation, the sun gear or output shaft revolves faster than the carrier or input shaft. Inasmuch as the spring clutch is attached to the sun gear and the latter revolves faster than the clutch's pocket or driving member *A*, the spring is unwound into a relaxed position to permit a disengagement or free wheeling between drive and driven shaft.

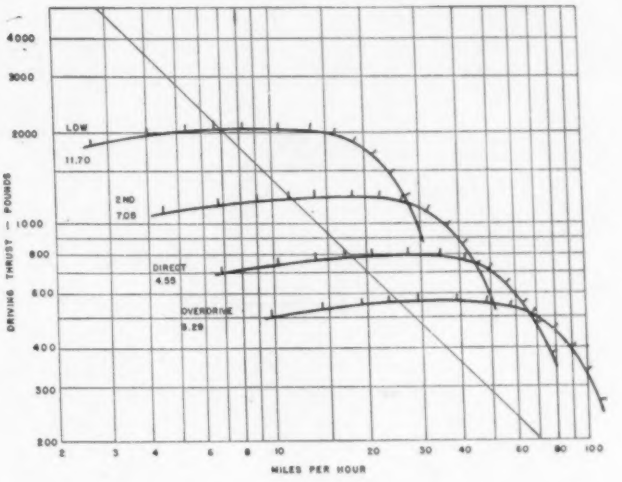
Now if the cone clutch can be made to disengage suddenly from the case, the speed of the output shaft would immediately drop and the entire drive line would be broken, were it not for the oneway clutch. As the clutch pocket or input shaft would attempt to revolve faster than the output shaft, the spring clutch will expand tighter to its clutching surface and lock the two members against relative rotation, making the planetary unit a direct drive. Obviously, a downshift has been completed without clutch feathering and without breaking the drive line.

Continued movement of the cone clutch merely locks the ring gear and carrier together to prevent free wheeling, which could happen during coasting when the output shaft tries to revolve faster than the input shaft.

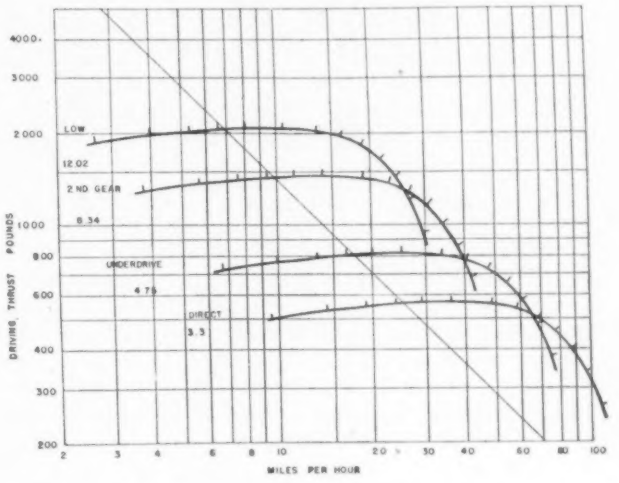
Now to effect an upshift under full torque load, the cone clutch is moved to engage the case smoothly. As the clutching surfaces between the carrier and ring gear are broken there is no relative movement between any of the planetary members because the spring clutch couples the input and output shafts so long as there is a driving load. As the cone clutch wedges against the case, the reaction member *R* comes smoothly to a stop in its attempt to start revolving in the opposite direction, it having been revolving at the same speed and in the same direction as the input shaft. Now, the output shaft starts revolving at a higher speed than the input and the unit is in overdrive ratio.

Thus, the upshift has been completed without overlap and without interrupting the input or output torque; and such a shift can be made with the engine at full throttle.

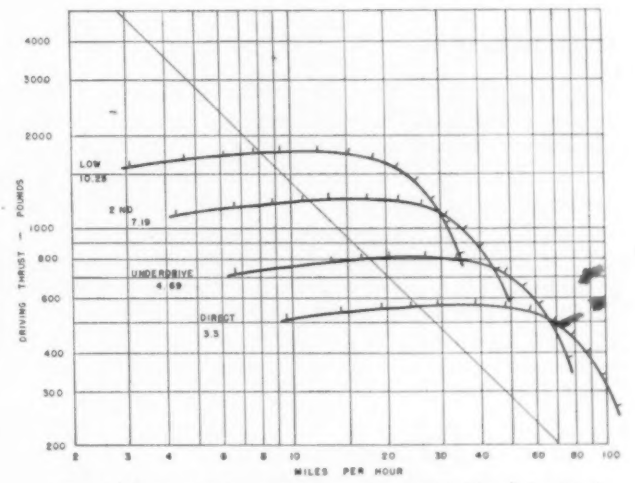
One again asks, why not two such units to make a successful 4-speed transmission, either full or semiauto-



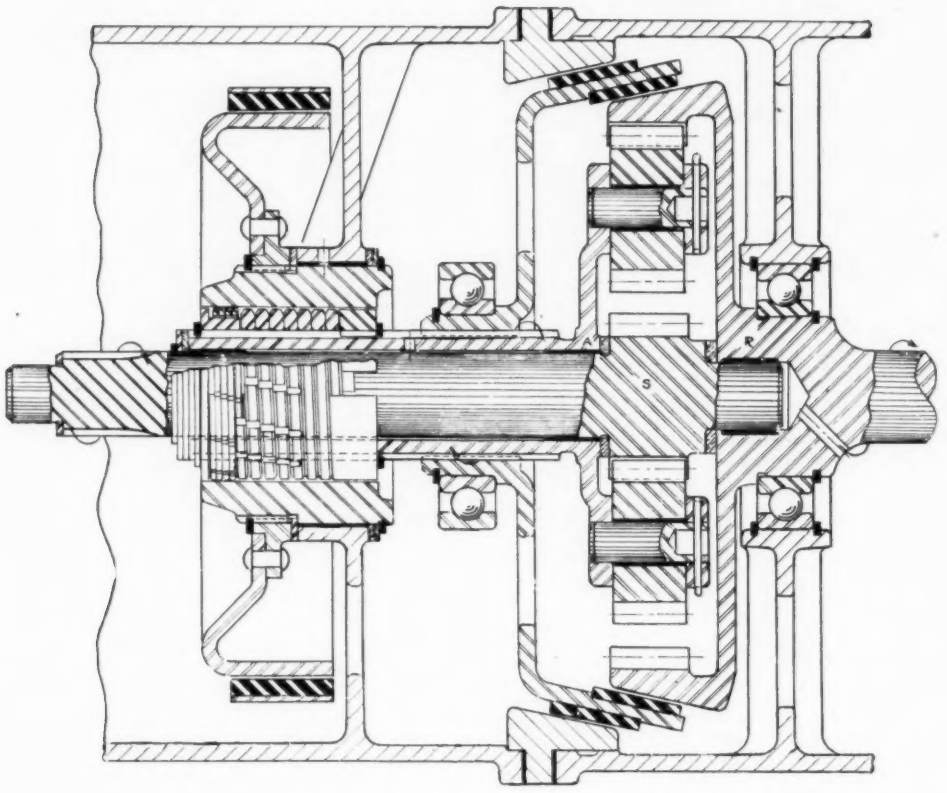
■ Fig. 4 – Three-speed transmission with semiautomatic overdrive



■ Fig. 5 – Four-speed automatic planetary transmission with overlapping clutches



■ Fig. 6 – Proposed 4-speed transmission shiftable under load



■ Fig. 7 – Underdrive with master clutch

matic, and one which will fulfil the conditions set forth in classification No. 4.

This would be ideal but for the fact that it is fundamentally incorrect to use overdrive where output shaft speeds are stepped up only to require stepping down again for final drive.

Obviously then, if the same principle is applied to underdrive and with proper hydraulics as the flexible means of control, one could have a fundamentally correct basis from which to start the design of a transmission that will shift under load.

■ Shifting Time

Shifting time in a step transmission is one requirement to be met which, if not at a minimum, will make the best automatic transmission a failure.

The shifts must take place in a minimum of time, for if the driver knows that a shift is to take place and he has time to wonder if it is going to be accomplished, then that time interval is too long. To shift at nearly the right moment in a minimum of elapsed time under every condition of temperature, speed, and torque conditions is a big undertaking. One can readily see that if overlap is used or a break in the drive line is required then the timing is much more difficult.

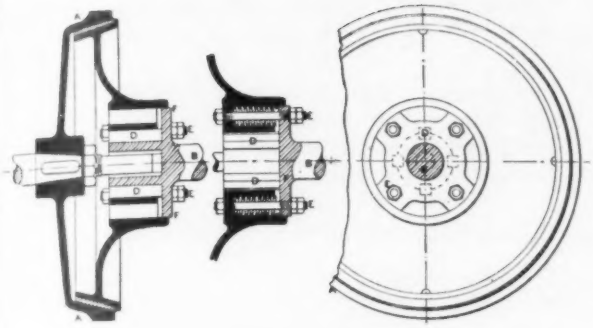
■ Basic Design Requirements

The author believes the following to be the basis from which to start the design of a full- or semiautomatic power shifted transmission as outlined in classification No. 4:

1. Four-speed progressive shift with direct drive in fourth.
2. All planetary gearing with step increase in ratios in the order of 35 to 45%.
3. Double-acting cone clutches, hydraulically operated by annular pistons, without piston rings.
4. No clutch pedal. Master or starting clutch within the transmission to be engaged by variable pressure dictated by throttle movement.
5. Make hand control lever optional, as it merely replaces the governor. (This for the driver who wishes to control the shifting.)
6. Valve block to be installed without the necessity of timing, making it easily replaceable in service.
7. Pressure lubrication to all bearings and no gears churning oil.
8. Main shaft mounted pump, front and rear, to eliminate gear drives. Use 2-stage front pump for high volume at low speed and cut out one stage automatically at higher speeds.
9. In the case of the full-automatic transmission it must be torque and speed responsive within reasonable limits.

■ Selecting Proper Ratios

Overall gear ratios on today's cars vary from 13.2 in first gear to 3.64 in top gear on one make and 10.8 in first to 3.29 in top on still another make. The main reason for the two extremes in first gear may be due to the general opinion that planetary gearing does not lend itself to flexible



■ Fig. 8 – Main clutch of Peugeot car (1896)

ratios; that is, where one desires certain inbetween ratios the remaining ratios are inherently altered.

There appears to be no basic reason, other than overall design dimensions, why a first-gear ratio need be higher than in the order of 10 to 1 for most applications. Also, dictated by current practice and dependent somewhat upon power-weight ratio, the top gear may be in the order of 3.3 to 1.

■ Torque Curve Comparison

Fig. 4 represents a torque curve, using the 3-speed with overdrive transmission in Fig. 1. The curve is plotted on a log-log sheet to show lb thrust at the wheels for a given mph, while the overall ratios indicate the relative positions of each step for the respective speeds. Note the wide step between first and second.

Fig. 5 represents the same torque curve, using the automatic transmission in Fig. 2, recalculated to represent the same top gear ratio in order to show comparison in steps. Note the wide step between the second and third.

Fig. 6 represents the same torque curve with adjustments made in gear ratios to make the first- to second-speed gap of Fig. 1 and the second- to third-gap of Fig. 2 more uniform with the third- to fourth-speed gap of both transmissions.

The overall purpose, of course, is to permit a more uniform step during a power shift, without overlap and without breaking the drive line, and to maintain ratios to which the majority of drivers are accustomed.

Many engineers are of the opinion that such a ratio spread is unimportant as long as there exists a smooth engagement during shifts, but it is evident that the clutches would have more work to do, and in the case of overlapping to complete the shifts where some 40 clutching surfaces must interchange loads, one can readily see where consistently accurate timing would be difficult.

Another important reason for a shorter gap between second and third is that such a shift requires a complete interchange of gear sets, and even where it is the type that will shift without breaking the drive line, the third-speed clutch would be required to pick up a greater load the greater the gap.

In short, during a shift from second to third, one gear set drops from direct drive to ratio simultaneously, with the other gear set being shifted from ratio to direct.

Obviously, if the transition is unbalanced or out of time due to too much overlap, the shifts may take place in the order of 1-2-4-3-4 during a complete upshift, while with insufficient overlap, which would be the equivalent of breaking the drive line, the shifts may take place in the order of 1-2-1-3-4.

The ultimate, of course, is a progressive shift of 1-2-3-4, which can be accomplished in a transmission where shifts take place without overlap requirement and without breaking the drive line, as outlined in classification No. 4.

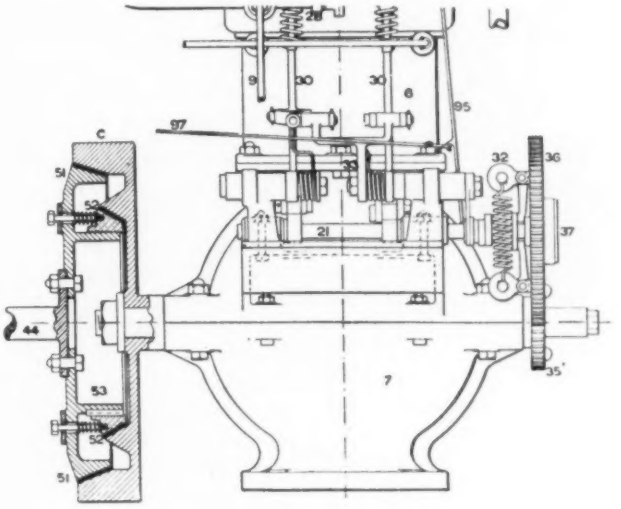
■ Planetary Gearing

The author believes that planetary gearing is the correct basis from which to start a transmission of the type in classification No. 4. With two gear sets in tandem and with the use of double-acting cone clutches similar to the DeNormanville, along with the master clutch attached to the reactionary member of the third gear set, it would result in:

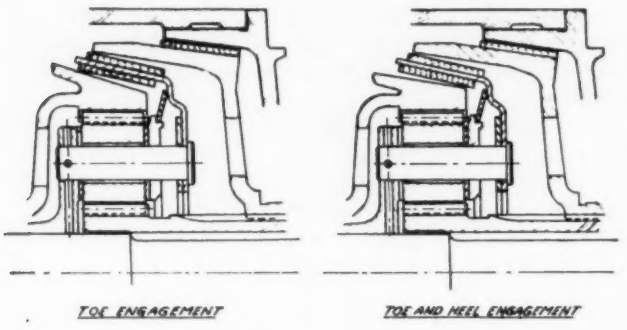
1. More or less pure radial gear and shaft loads.
2. Compact design for the gearing, pistons, and pumps.
3. Four available speeds using only two gear sets, with a minimum of gears.
4. Nesting of cone clutches.
5. No gears churning oil.
6. Relative simplicity in pressure lubrication to bearings.
7. An unbroken cylindrical case making itself impervious to distortion due to torque reaction.
8. Flexible planetary ratios in spite of views to the contrary.

■ Master Clutch

Used with the conventional type of transmission the master clutch is usually of the dry-disc type and must withstand full engine torque capacity. When used with planetary gearing it is no longer necessary to be of such capacity because the load reaction for a 1.38 to 1 underdrive planetary gear ratio is only 38% of input torque. Why then



■ Fig. 9 – Phoenix-Daimler engine – valve gear side and Panhard & Levassor clutch (1899)



■ Fig. 10 – Cone clutch illustrating feathering design and spring-loaded friction disc

is it not fundamentally correct to mount the starting or master clutch on the planetary reaction member as shown in Fig. 7? It goes without saying that the less load a clutch must handle the easier it is to make a feathering engagement.

Obviously, a transmission that can be shifted without breaking the drive line and without overlap should require no further master clutch engagement once the car is in motion.

In Fig. 7 a band type of clutch is shown on the reactionary member merely to indicate the master clutch function. As the driving member *A* revolves in the direction of the engine, the reaction member or sun gear *S* revolves in the opposite direction. In this direction the spring clutch expands to pick up the drum to revolve it in the same direction as the sun gear. As pressure is applied to the clutch band it brings the drum smoothly to a stop. The cone clutch then engages the case to prevent the output shaft from free wheeling during coasting. As the cone is brought into smooth engagement with the ring gear the reaction load is smoothly lifted from the oneway spring clutch, permitting the sun gear to revolve in the direction of the engine without the necessity of releasing the clutching band.

When the cone clutch is disengaged from its ring-gear contact, the reaction load drops immediately to the oneway spring clutch and the unit is again in underdrive ratio.

Thus a full upshift and downshift have been completed without overlap and without breaking the drive line, using the underdrive gear set.

■ Cone Clutches

No transmission is better than its clutches, and since cone clutches lend themselves to compact design a brief description of known facts about them is in order.

As most of us know, cone clutches are by no means new. Beaumont[2] describes the Peugeot 1896 car as having a cone master clutch. (See Fig. 8.) He describes also the Phoenix-Daimler of 1899 as having a feathering cone master clutch. (See Fig. 9.)

Pagé[3] has described a cone clutch of English design, operating in oil.

[2] See "Motor Vehicles and Motors," by W. W. Beaumont. Published by J. B. Lippincott Co., Philadelphia, 1902.

[3] See pp. 680-681 of "Modern Gasoline Automobile," by V. W. Pagé. Published by Norman W. Henley Publishing Co., New York, 1928.

Heldt has written much on cone clutches prior to 1920.

DeNormanville, in April, 1940, demonstrated to engineers in this country his overdrive incorporating a cone clutch.

Like planetary gearing, the cone clutch has come and gone and is apparently on its way back.

Strangely enough, *for compact design, one is strikingly dependent upon the other.*

■ Cone Clutches versus Other Types

The conventional dry disc clutch has not received recommendations for unit pressures in excess of 65 psi, and many designs use from 12 psi (for heavy duty) to 28 psi (for pleasure cars). They are operated dry because it makes for a simpler design, a higher coefficient of friction, and relatively easy service.

The cork insert clutch, with its oil bath, operates smoothly at proper oil level, mainly because the cork produces a high friction coefficient even in oil.

The multiple-disc clutch, with its bronze and steel discs operating in light oil, is apparently satisfactory even though special machinery was required before it could be made in production. While the friction coefficient is very low, it is compensated for by the use of many plates, hydraulically operated. Needless to say, the drag would be quite noticeable.

The fluid flywheel, with its sealed-in oil, permits a somewhat smoother clutch engagement with excessive slip at low speeds, but it can only operate in connection with a master clutch because it has no neutral.

The L.G.S. spring clutch provides a positive clutching engagement in one direction without slip. In short, it accomplishes the same results as does a jaw clutch without the necessity of full-speed crossing for engagement, but it may be energized or de-energized with very little effort while still under full torque load and will withstand a much greater load for a given size than clutches they replace.

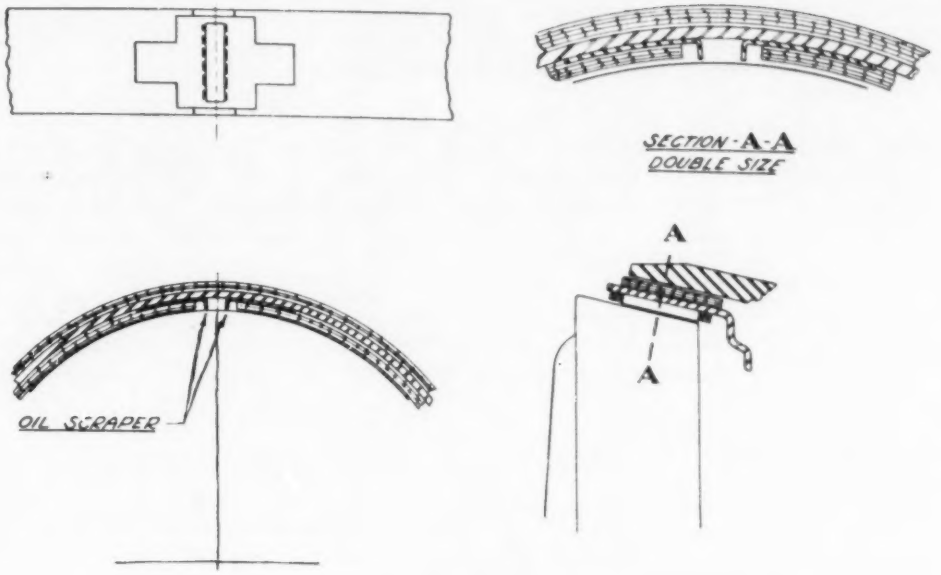
The cone clutch has a mechanical advantage due to its wedging action which indicates that it may be made smaller than the conventional dry disc type.

The cone lends itself to a compact, nesting design when used with planetary gearing, and even though it is made to operate in a mist of oil, the coefficient of friction does not drop as fast as the mechanical advantage gains.

Now with the introduction of cone clutches operating in an oil mist, a necessary increase in unit pressure is indicated. It is known that unit pressures up to 420 psi are used. It is apparent that since the clutch is to engage against an oil film, the pressure must be great in order to break through, and once it does, we again approach the dry clutch coefficient. Hence, in direct contradiction to the low unit pressure dictated in dry disc clutches, the turning point has been reached in wet clutches and high unit pressures appear necessary.

Carrying the design still further, by adding more friction members the clutch capacity is increased in direct proportion.

The double capacity cone clutch may be termed a planetary clutch and, as shown in Fig. 10, when clutched to lock up the gearing its capacity is increased in direct proportion to each friction face added, in this case, two friction faces.



■ Fig. 11 – Oil scraper design

Types of linings and methods of application: Many types of linings are available, among which are molded segments, split-ring molded, wire woven, and asbestos woven. All may be applied either by rivets or heatset with cement, or both.

Oil grooves, oil scrapers, and type of oil: The grooves may be any shape consistent with conventional manufacturing methods but the quantity of grooves and sizes must be determined by experience.

According to past experience the oil grooves will play a very important part in feathering as well as limit the engagement time as affected by friction coefficient changes. Such coefficient changes come about as the oil is squeezed out and the first contact between friction material and metal takes place.

Some types of oil scraper (see Fig. 11) may be required to remove excess oil from the drum prior to engagement. This will obviate the necessity of extremely high pressures to break through the oil film. In short, it appears that there should be a gradual, progressive increase of pressure until the bump is smoothed out, instead of a large pressure to break through the film, then a quick feathering pressure release, followed by a progressive pressure buildup to complete the engagement.

A light, nonvarnishing oil with rust inhibitor should be used.

Cone angles: If the tangent of the cone angle is less than the coefficient of friction the cone will stick.

At first glance one would suppose the above to be the limiting factor in a sticking cone during release or in feathering during engagement. It must be apparent, however, that sticking conditions may be further affected by softness of friction material, resiliency of certain metal parts, eccentricity of mating members, errors in coinciding angular faces, apparent vacuum grip between lining and drum, and many other things.

The 12-deg angle, based upon engineering review as well as current practice, seems to be the basis from which to start. It appears as if all tolerances in angular faces should be in the direction of a toe engagement (see Fig. 10), and in such magnitude as may be consistent with test results. It is even probable that the angle will have to be in error for toe engagement from the start. From observations to date, it would seem that a heel engagement would be wholly unsatisfactory in that full engagement would never be realized.

Lamination of either metal or friction material may be another solution to permit smooth engagement from a mechanical standpoint.

■ Hydraulic Application

Many methods can be employed for feathering engagement when clutches are actuated hydraulically, such as:

A pressure overlap valve may be used when the ratio change is accomplished by an interchange of gear sets. Such a valve is designed to operate only when pressure builds up to a predetermined figure, based on the resistance of the clutch moving into engagement. It may be well to note that such a design functions consistently irrespective of lining wear.

Restricted bleed may be used on one side of the piston when pressure is admitted to opposite side.

A flood valve will permit the chamber which is to receive pressure to remain full during operation, so that approximately the same amount of oil is used during each shift.

A variable pressure valve as dictated by throttle movement is usually necessary when the clutch is used as a primary or starting clutch. This is apparently less necessary when initial engagement is made with a fluid coupling or a torque converter before the accelerator is depressed.

Restricted pressure may be desirable in some instances where purely a time element is the deciding factor for feathering. This is similar to restricted bleed but is a function of the pressure side.

Pump pressure may require variation with throttle movement or with shift lever position to offer momentary high pressures under certain shift conditions.

Timing in the sense of an adjustable unit is not even being considered. If necessary, however, it may be accomplished by using a two-piece slip-link valve or, as stated above, a pressure overlap valve. Neither is dependent upon lining wear.

A belleville spring or its equivalent, as shown in the double clutch (Fig. 10), apparently has its place in that initial engagement at light throttle will be taken by one facing, while during a slam engagement the spring will unload one facing by the amount of the spring's total load.

The process of elimination, as determined by tests, is accepted as one of the major problems of experimental work. Without a doubt, many interesting surprises await us, but these findings, coupled with the knowledge of engineers closely connected with transmission development and testing, should provide a good start on producing a satisfactory cone clutch application in the shortest possible time.

■ Conclusion

The author believes that a power shifted, full- or semiautomatic planetary transmission, using hydraulically operated cone clutches, can be made without deviating appreciably from ideas already in use or on the shelf.

There is hardly a transmission company that is not familiar with the items listed herein and that, coupled with a few extras that have nearly passed their patent life, should give us the new combination of old ideas that will result in a successful automatic transmission.

SIGNIFICANCE of CETANE NUMBER in FUELS

continued from page 439

range of techniques available, such that diesel fuel of practically any desired composition can be produced, provided there is sufficient economic incentive. If the range of processes, such as catalytic and thermal cracking, isomerization, hydrogenation, dehydrogenation, solvent extraction, cannot produce hydrocarbon mixtures of the desired composition from petroleum stocks, then recourse can be had, as in the aviation industry, to synthesis of the desired components.

One of the most interesting of the synthesis processes is the Fischer-Tropsch, put into large-scale production by the Germans. In the Fischer-Tropsch synthesis a mixture of carbon monoxide and hydrogen derived from coke, coal, or natural gas is reacted over a catalyst. By varying catalyst, feed composition, operating pressure, and temperature, oils of different characteristics can be produced. The product from cobalt catalyst is composed essentially of straight-chain paraffins. A typical inspection of diesel fuel obtained from a cobalt catalyst is given in Table 3. It will be noted that this stock has a cetane number of 90 and is essentially sulfur free.

By use of an iron catalyst, a fuel composed mainly of mono-olefines along with some aromatics and naphthenes is obtained. Recent data obtained from Germany have shown that the process can be modified to produce high yields of isoparaffins. Large-scale commercial application of the Fischer-Tropsch principle in America is very close to reality. By application of the fluid catalyst technique and locating the plant close to the source of large supplies of cheap natural gas, costs of the process may be reduced to the point where high cetane diesel fuels can be made at costs competitive with similar quality fuels made from petroleum crude.

■ Conclusions

1. In high-speed diesel engines of present design a certain minimum cetane number is required to ensure satisfactory starting and to minimize smoking, combustion roughness, and engine deposits.

2. Cetane number is a measure of ignition quality of a fuel. There is evidence that other characteristics indicative of the rate of flame propagation or its burning properties during the combustion phase are of equal importance in diesel-engine performance.

3. The use of very high cetane fuels has shown no great advantages in engines of present design. No data have been obtained using engines especially designed to take advantage of this property. However, it is believed that marked increase in engine power output can be obtained by use of high cetane rapid burning fuels in modified engines of reduced compression ratio with a high degree of supercharge.

Table 3 – Inspection on Cobalt Catalyst Fischer-Tropsch Diesel Oil

Gravity, deg API	51.7
Flash, PMCC, F	192
Sulfur, %	0.002
Ash, %	Nil
Color, ASTM	0
Copper Strip Corrosion at 212 F	Pass
Pour Point, F	30
Viscosity, SU at 100 F, sec	34.1
Diesel Index	98
Cetane Number	90
Conradson Carbon on 10% btms, %	0.08
BS&W	Nil
Olefins, %	4
ASTM Distillation, F	
IBP	412
5%	439
10%	445
20%	458
30%	468
40%	482
50%	495
60%	513
70%	534
80%	558
90%	585
95%	600
FBP	619
Recovery	98
Residue	1

4. Study of data on ignition quality of pure hydrocarbons and on additives indicates that fuels of widely varying composition and combustion characteristics with the same cetane number can be produced.

5. Basic combustion research is necessary to determine the optimum composition of diesel fuels required for high-output engines.

6. Modern refining techniques, developed through stimulus of gasoline research, offer great flexibility for production of diesel fuels of special composition, provided sufficient economic incentive is established.

Combustion in a

PRECOMBUSTION-TYPE DIESEL ENGINE

by HARRY F. BRYAN
International Harvester Co.

IN 1938 a high-speed motion picture study of diesel-engine combustion was started by the International Harvester Co. in their Gas Power Engineering Department in Chicago. The investigation has been carried on in two phases since its inception: (1) a visual and photographic study of spray form, spray penetration, and combustion in a pressure chamber or bomb in which engine conditions of pressure and temperature can be simulated; (2) a photographic study of combustion in a full-scale single-cylinder diesel engine through windows set in the precombustion chamber.

The high-speed camera used for this work is of the optical compensator type made by the Eastman Kodak Co. When purchased the maximum safe speed was 2000 frames per sec. Improvements suggested by the maker and the Harvester Instrument Department made this camera dependable at 2500 frames per sec, and short films can be made at 2800 frames per sec. At 2500 frames per sec, 50 ft of 16-mm film is exposed in 0.8 sec. Although purchased for a specific purpose this camera is now considered routine test equipment.

Apparatus

It may be well, first, to describe some of the collateral equipment necessary for taking these pictures, since the design and construction of this equipment is quite a project in itself and in most cases determines the success or failure of the program.

A pressure chamber developed to study injection nozzle spray characteristics and combustion under simulated engine conditions is shown in section in Fig. 1. Electric heaters in the walls of this chamber make it possible to elevate the air temperature in the bomb to 600 or 700 F. Compression pressure up to 1000 psi is supplied from a tank of compressed air and the required pressure maintained with an automatic balance valve.

When this chamber is filled with nitrogen or other inert gas, fuel sprays discharged into it can be studied at high temperature and pressure without combustion. Self-ignition and combustion of the spray can be studied when the chamber is filled with air and the air temperature and pressure raised to approximate engine operating conditions.

Two windows 2¾ in. wide, 5 in. long, and 1 in. thick are located on opposite sides of the chamber. There is also a window 2¾ in. in diameter on the end of the chamber opposite the nozzle. The windows are made of "tuflex" glass. A bank of lights back of the rear window throws the spray into silhouette as it passes between the large windows. The total volume of the chamber is 94 cu in.

The complete apparatus for photographic study of the sprays in this chamber is shown in Fig. 2; from right to left in this view is the camera, pressure chamber, and bank of lights. Under the bench is the speed control for the camera, and the oscillator for the timing spark on the film.

A full-scale single-cylinder engine with 4¾-in. bore and 6½-in. stroke is used for combustion study in the engine. This engine with the camera mounted rigidly in place is shown in Fig. 3. The windows in the precombustion chamber are normal to the optical axis of the camera lens and a lamp with just sufficient wattage to throw the spray in silhouette is located back of the rear window.

Since the engine runs only a few hundred revolutions during the time necessary for the film to be exposed, it

[This paper was presented at the SAE Annual Meeting, Detroit, Jan. 16, 1946.]

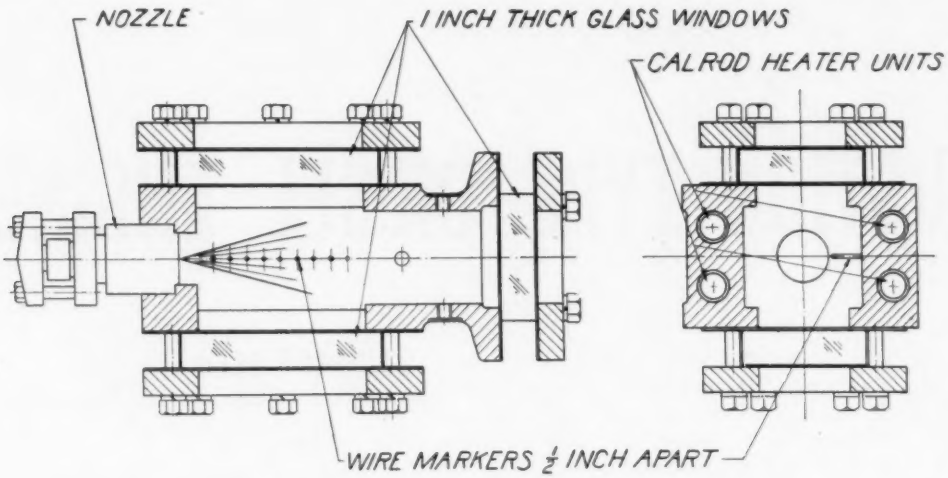
A HIGH-SPEED motion picture study of diesel-engine combustion has been made along the following lines:

1. A visual and photographic study of spray form, spray penetration, and combustion in a pressure chamber or bomb in which engine conditions of pressure and temperature can be simulated.

2. A photographic study of combustion in a full-scale single-cylinder diesel engine through windows set in the precombustion chamber.

The apparatus used in this investigation as well as the results achieved are discussed in full by Mr. Bryan.

THE AUTHOR: HARRY F. BRYAN (M '24) is now a research engineer in the Mechanical and Development Division of the International Harvester Co., Chicago. Before joining International Harvester in 1927, he was assistant chief engineer of the Ensign Carburetor Co. of Los Angeles. During World War I he served in the American Tank Corps where he first became interested in the carburetion problems of trucks and tractors. Mr. Bryan has been active in SAE Chicago Section and has served as Section chairman and also as treasurer for three years.

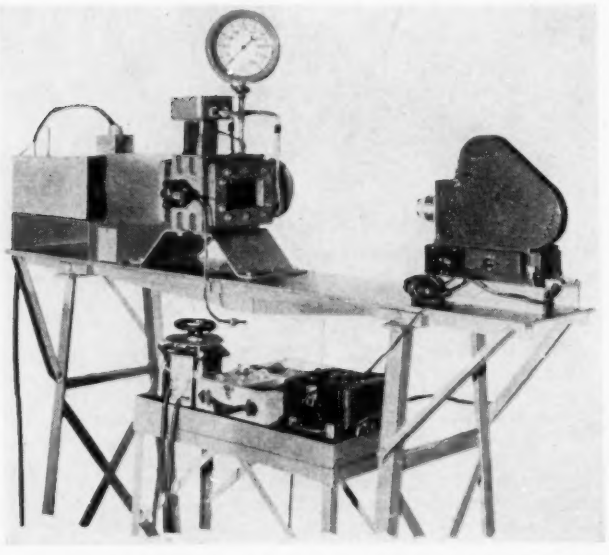


■ Fig. 1 – Pressure chamber for spray study

must be kept at operating temperature by external means. A heater is attached to the cooling system of the engine and heated glycerine is circulated through the engine to maintain the required temperature. The addition of a special cylinder head with a cooler and bypass in the system makes it possible to raise and lower the temperature of the precombustion chamber and cylinder head independently of the cylinder wall temperature. The heater is shown at the right of the engine in Fig. 3.

When indicator cards are to be taken simultaneously with the combustion photographs, a piezoelectric pickup is mounted in either precombustion or main combustion chamber. The voltage impulse from this pickup is amplified and carried to the vertical plates of a cathode ray oscilloscope and the horizontal sweep is controlled by a contactor operating at one-half engine speed. A second camera also driven by the engine crankshaft is synchronized to record the trace on the oscilloscope screen. This apparatus is shown in Fig. 4.

Type of Engine – The diesel engines manufactured by this company are all of the 4-cycle precombustion-chamber type. The precombustion chamber shown in Fig. 5 was developed in the Gas Power Engineering Department and has given very satisfactory performance. This type of combustion chamber is comparatively insensitive to normal variations in the physical properties of the fuel, and for this reason a very simple and rugged injection nozzle construction can be used. A heat resisting liner in the chamber is dimensioned to maintain the required operating wall temperature and the angle and area of the throat in this liner controls the amount of turbulence in the chamber.



■ Fig. 2 – Apparatus for photographic study of diesel sprays

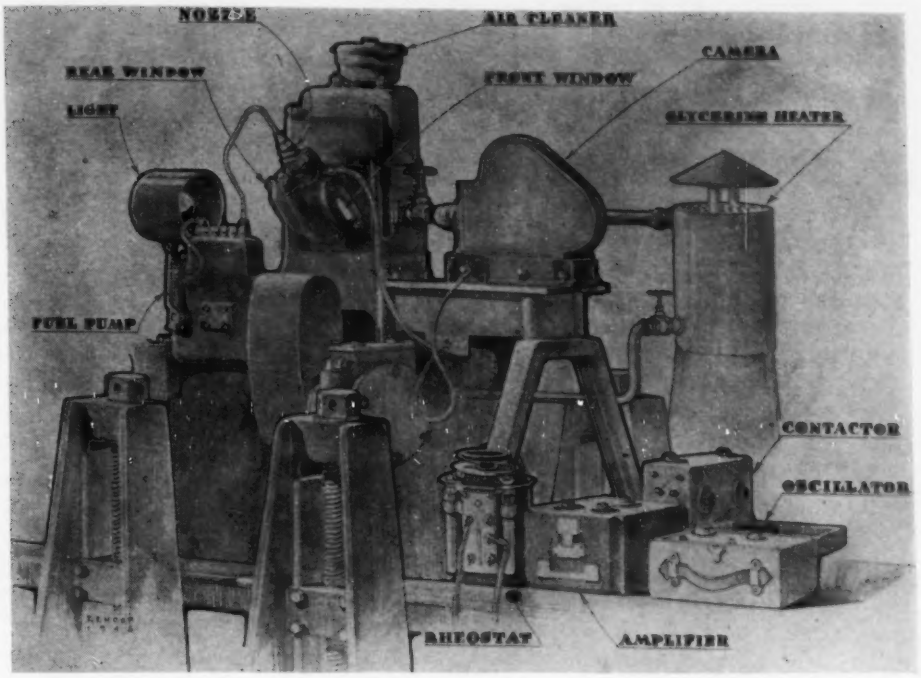
The elliptical shaped windows through which combustion is observed in the single-cylinder test engine are on opposite sides of the chamber with their long axis parallel to the axis of the precombustion chamber. The windows are 13/16 in. wide and 1 15/16 in. long and extend from the nozzle to just above the throat of the chamber.

Combustion Characteristics Discussed – Combustion control at the low speed end of the torque curve in an automotive-type diesel engine is generally a rather difficult problem. This is particularly true in tractor and other industrial powerplant installations where high torque at low engine speed is required. This may be due to a variety of causes, such as early start of injection, which has been set for optimum performance at rated speed, coarse sprays, over-penetration, or reduced combustion-chamber wall temperature.

This paper will be confined to a discussion of some of the characteristics of the precombustion-chamber type of diesel engine under full load at 550 rpm to 600 rpm. This engine has a normal governed speed of 1300 to 1400 rpm, and pump timing is set for optimum full-load performance at 1300 rpm.

Combustion in the precombustion-chamber type of engine proceeds in an entirely different manner from combustion in the spark-ignition engine. In the latter, when operating without detonation, the flame progresses in an orderly manner through the mixture and shows a marked dividing line between the burned and unburned gases throughout the combustion cycle. The rate of burning is comparatively low.

On the other hand, combustion in the precombustion-chamber type of diesel engine follows the turbulent motion of the fuel spray and air and it is not possible to define



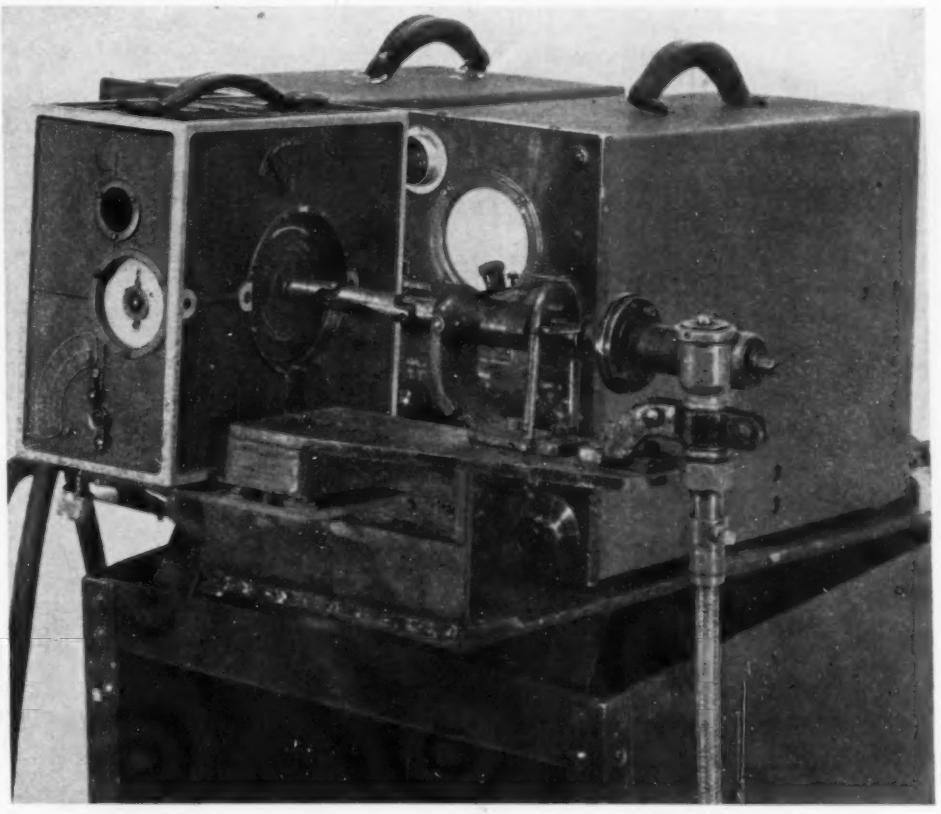
▪ Fig. 3 – Single-cylinder test engine for photographic study of diesel combustion

clearly the boundaries of the burned and unburned gases once the mass of fuel and air is ignited.

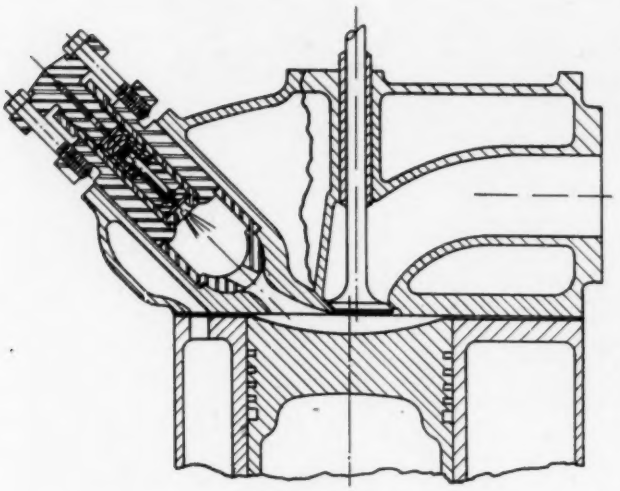
However, this does not mean that the diesel engineer can disregard the excellent work on combustion control that has been done by the spark-ignition engine engineers.

Combustion started by compression ignition in a quiescent, constant-volume chamber, such as shown in Fig. 1, is quite similar to spark-ignition engine combustion in that it spreads at a comparatively uniform rate through the combustible mixture. A study of the fuel spray and combustion of the fuel spray in the pressure chamber under simulated engine pressure and temperature conditions as a preliminary to a study of combustion in the engine is essential.

Combustion in the Bomb – The enlargements of consecutive spray photographs in Fig. 6 show the formation



▪ Fig. 4 – Oscilloscope and camera used for pressure-time records



■ Fig. 5 – Combustion chamber for diesel engine

and penetration of spray from the nozzle when injected into the pressure chamber or bomb filled with nitrogen. Temperature, pressure, and other conditions of the test are given in Table 1.

Although the spray is injected into very hot gas the silhouette is heavy and well defined during the first four frames, and it is not completely vaporized after 28 frames or 90-deg crank angle. These pictures show clearly that both turbulence and the heat of combustion must be added to this spray to reduce the vaporizing time.

When the chamber is filled with air under the same test conditions, self-ignition and combustion of the spray occurs as in Fig. 7. Ignition occurs in the seventh frame (14.4-deg crank angle) after the start of injection. Combustion starts in the spray envelope about 1¾ in. from the nozzle. The flame very quickly surrounds the spray from the point of ignition to the spray tip but does not spread toward the base of the spray until a second ignition occurs at the nozzle, three frames later. The spray is completely enveloped in flame in the eleventh frame (28.6-deg crank angle), after actual start of injection. Burning continues for 33 frames. The total burning time is more than 120-deg crank angle. This is three times the duration of burning that can be tolerated in the engine.

The quick envelopment of the spray by the flame effectively cuts off the spray core from access to the surrounding air and definitely delays combustion. This is characteristic of all fuel sprays unless turbulent air movement is maintained to strip the envelope of fine fuel particles from the spray core and diffuse the fuel vapors into the air as fast as they form.

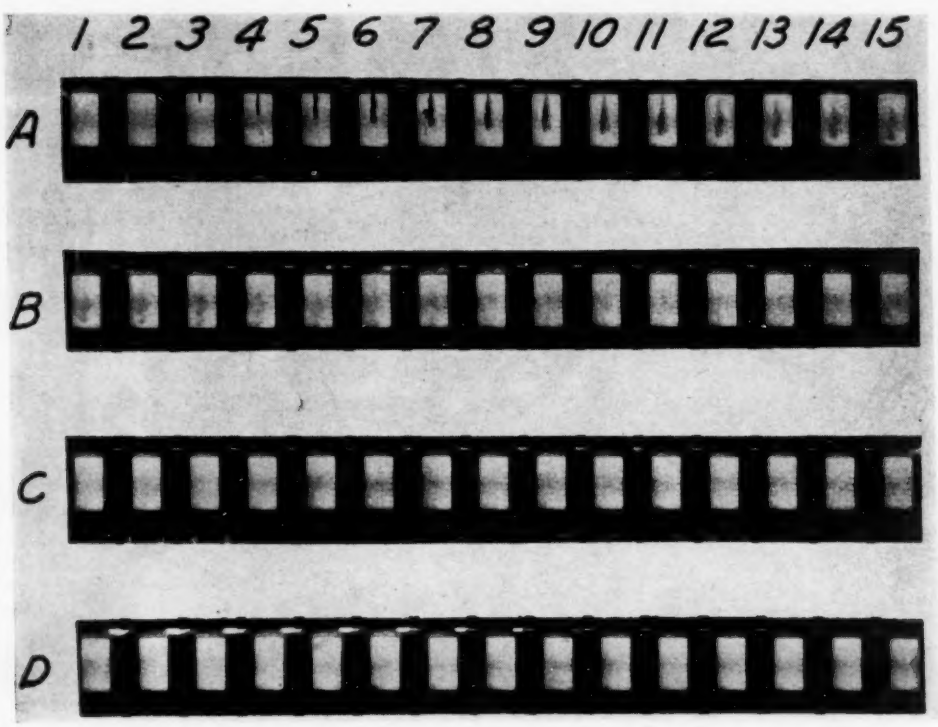
Table 1 – Conditions of Test

Injection Pump Speed, rpm	600
Nozzle Opening Pressure, psi	700
Gas in Chamber	Nitrogen
Temperature of Gas, F(abs)	1085
Pressure of Gas, psi	480
Camera Speed, frames per sec	2240
Pump Rotation, deg per frame	1.608
Crank Angle, deg per frame	3.216
Duration of Injection, deg pump rotation	11.3

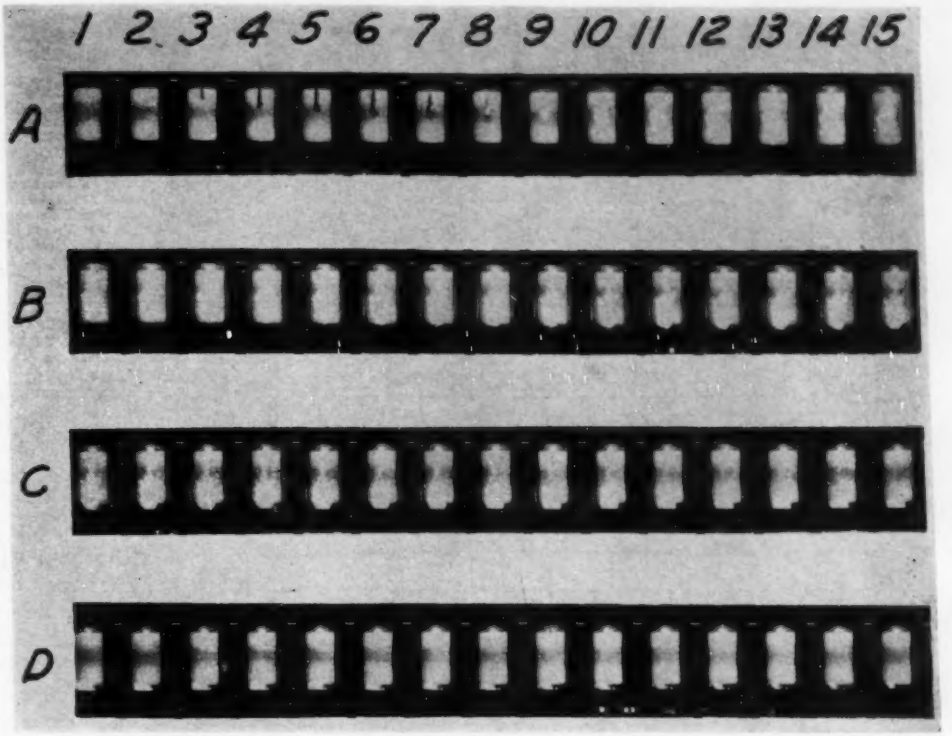
Combustion proceeds at approximately constant pressure because of the large ratio of air to fuel in the chamber. The rate of flame propagation is comparatively slow. No signs of the intense luminescence usually associated with diesel combustion can be observed. Combustion is practically smokeless.

Combustion in the Engine – Table 2 shows the engine operating conditions under which the next test was made.

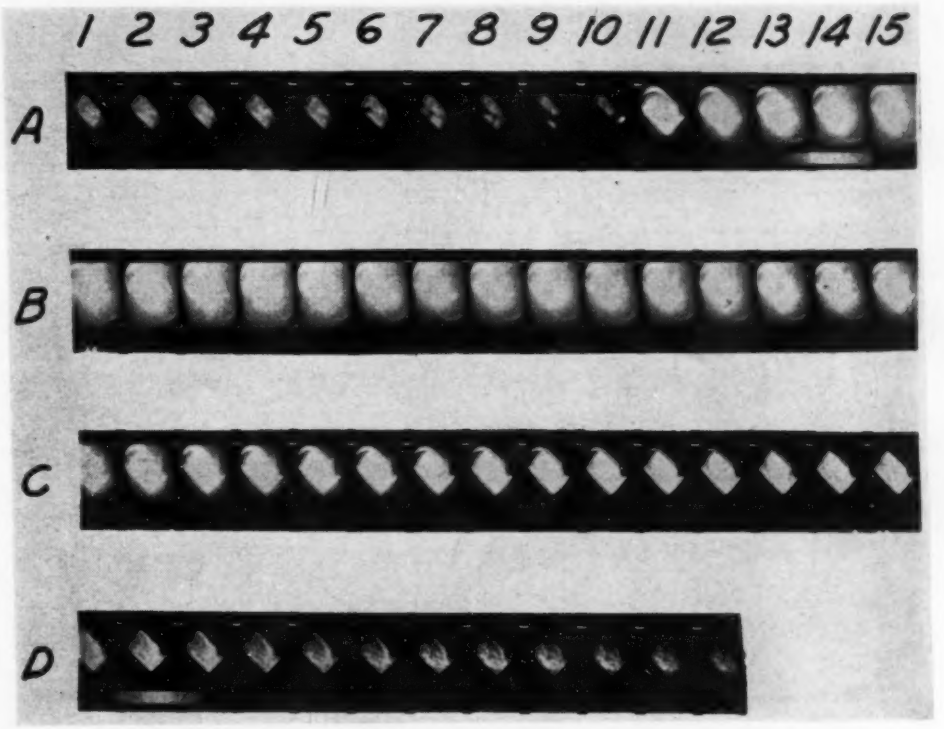
Actual combustion in the engine under this set of conditions is shown in Fig. 8. Note that the revolving mass of air in the precup has sufficient force to push the spray aside as it leaves the nozzle (frames 1-9). This effectively



■ Fig. 6 – High-speed photographs of spray from diesel nozzle in closed chamber



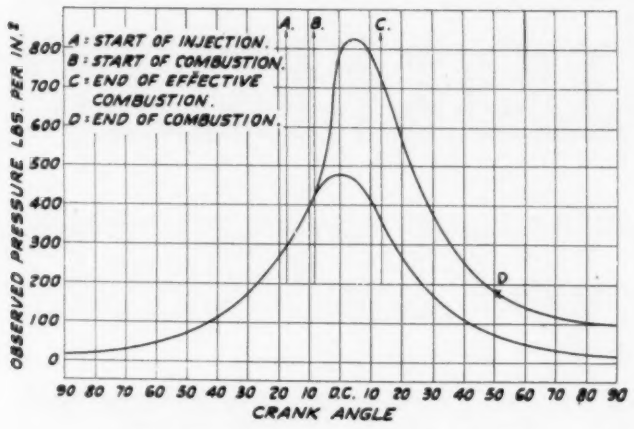
■ Fig. 7 – High-speed photographs of combustion of spray from diesel nozzle in closed chamber



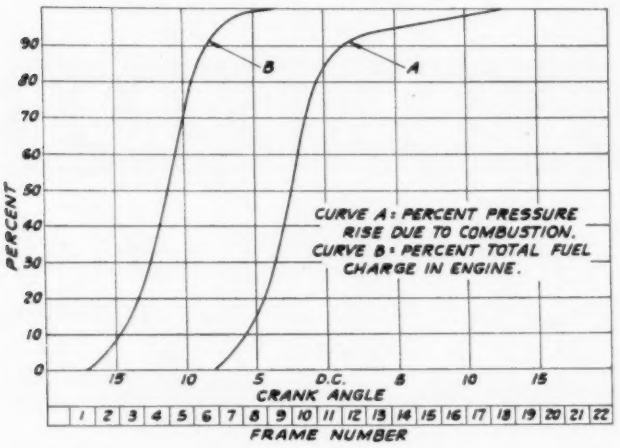
■ Fig. 8 – High-speed photographs of combustion in hot engine

strips the spray envelope from the spray core and carries it around the chamber. Five frames (8.8-deg crank angle) after the start of injection both the spray envelope vapors and the spray tip have reached the lower end of the precup and combustion starts in the vapors at this point. At top dead center, five frames (8.8-deg crank angle) later, the air and fuel in the precup become a flaming mass revolving at high speed in which a faint outline of the spray form can be observed. Six or seven frames (10.5-deg to 12.5-deg crank angle) after top dead center the mass stops revolving; it is at this point the piston motion is reducing the pressure faster than the liberation of heat can maintain it. Burning continues at a lower rate for eight or nine frames. The duration of combustion in the engine in terms of crank angle degrees is about one-third that observed in the bomb.

Considerable smoke forms during this combustion and is visible as soon as the smoke particles drop below the incandescent temperature. The intense luminosity of the



■ Fig. 9 – Firing pressure card from hot engine



■ Fig. 10 – Per cent pressure rise due to combustion in hot engine

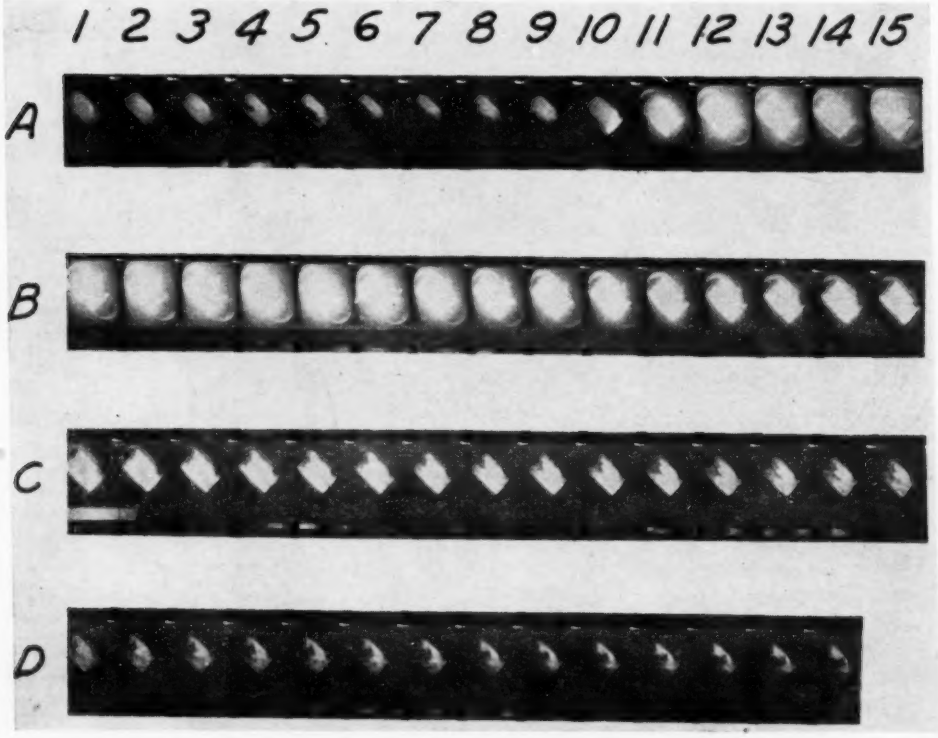
Table 2 – Engine Operating Conditions

Engine Speed, rpm	595
Nozzle Opening Pressure, psi	700
Compression Ratio	13.67:1
Injection Pump Port Closing, deg BTDC	23.5
Actual Start of Injection, deg BTDC	17.6
Actual Start of Ignition, deg BTDC	6.83
Entering Air Temperature, F	115
Temperature at Nozzle, F	190
Precup Wall Temperature—Top, F	335
Precup Wall Temperature—Bottom, F	330
Cylinder Wall Temperature, F	255
Crank Angle, deg per frame	1.76

flame is caused largely by the incandescent particles of carbon. Data from previous combustion studies show that the point in the chamber at which combustion starts has a great influence on smoke. When ignition takes place at the throat of the precombustion chamber, the rapid expansion of the hot gases forces them through the throat into the main chamber. These gases act as a separating wall between the rich unburned mixture in the precombustion chamber and the fresh air in the main chamber. As a result, combustion in the precombustion chamber proceeds in a very rich mixture of fuel and air, dissociation takes place, and smoke is formed. The fuel in the precombustion chamber must have free access to fresh air during combustion if smoke is to be controlled.

Although this picture of diesel-engine combustion is quite different from the orderly combustion which has been observed in spark-ignition engines when operating without detonation, nevertheless the characteristics of consecutive explosions in this engine will resemble each other quite closely, provided certain factors, such as rate of injection, spray texture, and precombustion-chamber wall temperature, can be held under control.

Indicator cards showing the pressure changes in the precombustion chamber were made at the same time combustion was photographed. The compression and combus-



■ Fig. 11 – High-speed photographs of combustion in cold engine

tion portion of the pressure-time curve for the preceding explosion appears in Fig. 9.

The correlation between indicator card and combustion photographs is fairly good. Ignition is followed by a high rate of pressure rise corresponding to the rapid increase in flame luminosity in precombustion photographs. The end of effective combustion is approximately 13 deg (crank angle) after top dead center; this corresponds to the point in which the flaming mass became stationary in the photographs. The slope of the expansion curve indicates continued burning at a reduced rate for some time. The curve itself is conventional and quite representative of pressure cards taken at low torque with early injection timing, but the rate of pressure rise and maximum pressure are too high for smooth operation at this speed.

Whenever a pressure card of this form is to be analyzed, the assumption is usually made that combustion takes place at constant volume because of the very small amount of piston movement during combustion. In the excellent work done by Rassweiler and Withrow[1] it was shown that the effect of piston motion on the pressure rise in an engine cannot be neglected, because "as the pressure level is increased, the pressure change produced by a given change in total combustion chamber volume also increases." The effect is even greater in the diesel because of the small combustion volume in these engines.

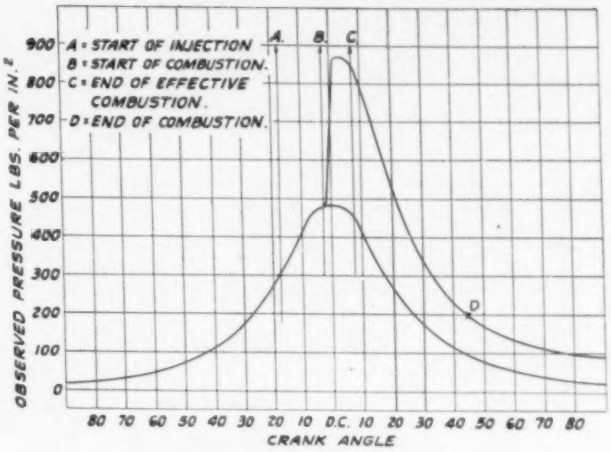
A method of sorting out the pressure rise due to combustion and the pressure rise due to piston motion from the total pressure on the pressure card was then proposed by the writers. After a careful study of the results obtained with this method of analysis they came to the conclusion that per cent of mass (fuel and air) burned at any instant in a normal engine explosion can be determined with fair certainty by finding the per cent pressure rise due to combustion at any instant obtained from the indicator card. This work was done on spark-ignition engines.

Although we do not have conclusive evidence that this method can be applied to all diesel engines to determine with certainty the rate of mass (air and fuel) burned under all conditions, we have found that the per cent pressure rise due to combustion taken from a diesel pressure-time card does help to evaluate the performance of the engine and can be used as a basis of comparison with other engines, particularly when a study of combustion control is being made.

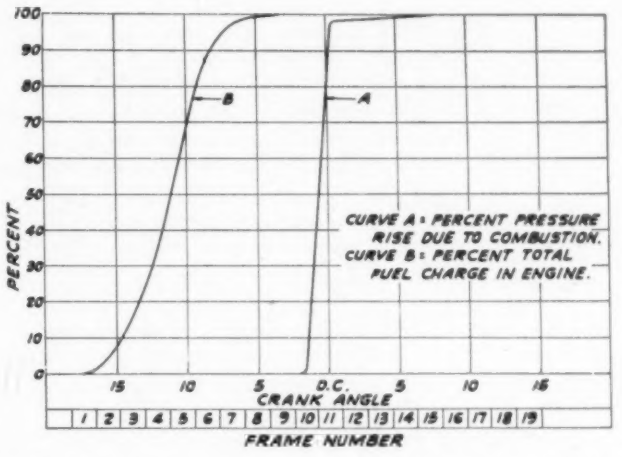
Plotted against crank angle curve *A* (Fig. 10) are the values for per cent pressure rise due to combustion as obtained by this method from the observed pressure-time curve in Fig. 9. The rate of pressure rise due to combustion at any instant of the combustion period can be obtained from the slope of this curve. In this curve the maximum rate of pressure rise occurs 6¼ deg (crank angle) after ignition and 82% of the total pressure rise is attained at top dead center.

Now if per cent of the total fuel charge present in the engine at any instant during injection is plotted in its observed relation to top dead center, as in curve *B* (Fig. 10); and if the assumption is made that curve *A* represents the per cent of mass (mixture fuel and air) burned at any instant during combustion, it is possible to make a direct comparison between the energy available in the combustion chamber at any instant and the rate of energy liberation by combustion. The degrees of crank angle separating the two curves at any point show the ignition lag of the fuel at that point.

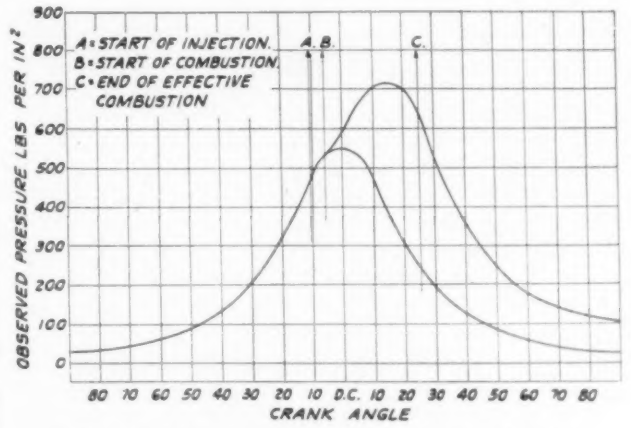
The curves *A* and *B* in this case are approximately


A = START OF INJECTION
B = START OF COMBUSTION.
C = END OF EFFECTIVE COMBUSTION.
D = END OF COMBUSTION.
OBSERVED PRESSURE LBS. PER IN.²
CRANK ANGLE

■ Fig. 12 – Firing pressure card from cold engine

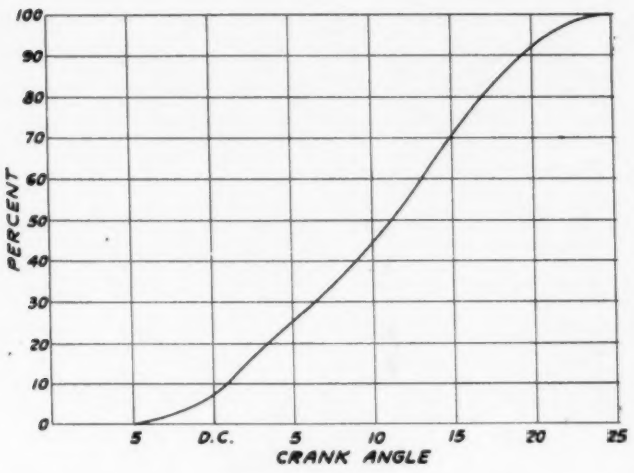

CURVE A = PERCENT PRESSURE RISE DUE TO COMBUSTION.
CURVE B = PERCENT TOTAL FUEL CHARGE IN ENGINE.
PERCENT
CRANK ANGLE
FRAME NUMBER

■ Fig. 13 – Per cent pressure rise due to combustion in cold engine



■ Fig. 14 – Firing pressure card from experimental engine

[1] See SAE Transactions, Vol. 33, May, 1938, pp. 185-204: "Motion Pictures of Engine Flames Correlated with Pressure Cards," by G. M. Rassweiler and L. Withrow.



■ Fig. 15 – Per cent pressure rise due to combustion in experimental engine

parallel. This would indicate that, although ignition occurs 8.8-deg crank angle after the start of injection, the fuel is vaporized, diffused into the air, and burned at the same rate that it was discharged into the combustion chamber. This is truly interesting when it is considered that 86% of the total charge of fuel is in the chamber at the time of ignition. However, the combustion photographs showed the outline of the unburned spray in the flaming mass for a number of frames after ignition.

The similarity of the two curves *A* and *B* (Fig. 10) is probably due to the fact that the precombustion-chamber wall temperatures had been adjusted for the best possible performance under the test conditions of injection timing, fuel quantity, spray form, and rate of injection.

The effect of reduced engine temperature on combustion in this engine is interesting. The engine temperatures were reduced to the values shown in Table 3 and another group of photographs taken.

The flame pictures in Fig. 11 show combustion under these operating conditions. The ignition lag has increased from 8.8- to 15-deg crank angle; two frames (3.5 deg) after ignition the entire mass of fuel and air is aflame. The total duration of combustion is 19 frames (33.5 deg) a little more than two-thirds the duration observed in the hot engine. Also, there was considerably less smoke formed.

The pressure-time card for this combustion (plotted in Fig. 12) also shows what is going on in the combustion chamber. Ignition occurs less than two crank angle degrees before top dead center and is followed by almost vertical pressure rise. The original oscilloscope trace of this curve, although highly damped, indicated high pressure fluctuations were present in the expansion curve, with peaks approximately 3.3-deg crank angle apart and of unknown height.

The per cent pressure rise due to combustion obtained from the pressure-time card (Fig. 12) is plotted as curve *A* in Fig. 13. The slope of this curve is almost vertical and now has no resemblance to the per cent of fuel charge present in the engine shown as curve *B* (Fig. 13). When the slope of the two curves is compared, it is found that although the ignition lag at the start of combustion is 15-deg crank angle it decreases as burning progresses until

Table 3 – Engine Temperatures

Engine Speed, rpm	600
Entering Air Temperature, F	80
Nozzle Temperature, F	136
Precup Wall Temperature—Top, F	115
Precup Wall Temperature—Bottom, F	180
Cylinder Wall Temperature, F	110

at 82% of the pressure rise the ignition lag is practically identical to that observed at 82% pressure rise in the hot engine. It is also interesting to note that pressure rise curves in both Fig. 10 and Fig. 13 cross top dead center at 82% of the total pressure rise. It is somewhat doubtful whether or not the curve *A* now accurately represents the rate of mass burning because of the high heat loss associated with this type of combustion. However, from the sound of the engine it is obvious that this curve does represent a rate of burning you do not want in a production engine.

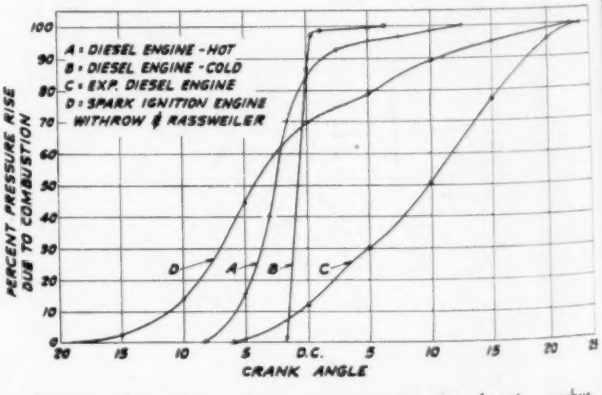
■ General Discussion

The curves *A* and *B* in Fig. 10 show that it is possible to have the rate of burning equal to the rate of fuel injection. In this case, however, the rate of burning is too high for smooth performance. To alleviate this condition either the rate of fuel injection or the rate of burning must be reduced.

The curve *B* in Fig. 10 is more or less representative of the rate of fuel injection from injection systems with multicylinder pumps and comparatively long injection pipes. Some variation in the duration of injection can be obtained in these systems by changing pump plunger size, injection line volumes, pump cams, opening pressures, and the like, but the general shape of the curve will remain approximately the same because of the effect of fuel compressibility.

In spite of the limitations, this type of fuel injection system has many excellent features which the engineer would like to retain. The data show that the form, relative volume, throat area, and other design factors in the precombustion-chamber system exert a greater influence on combustion than small variations in rate of injection.

The pressure-time card in Fig. 14 was taken from an experimental multicylinder engine with the same bore and stroke as the test engine and shows that improved performance can be obtained at low-speed full-load operation by giving careful attention to details of precombustion-



■ Fig. 16 – Comparison of per cent pressure rise due to combustion in diesel and spark-ignition engines

chamber design. Very little change in the rate of fuel injection was made to accomplish these results, although the duration of injection was increased, due to the increased amount of fuel that could be burned in the chamber.

This indicator card was taken at an engine speed of 600 rpm. The ignition lag is 4½-deg crank angle, ignition occurs approximately 5½ deg before top dead center, and effective combustion continues to 25 deg after top dead center. The maximum rate of pressure rise during combustion is no greater than the maximum rate of pressure rise of compression. The engine operation is smooth and comparatively free from smoke.

The curve of per cent pressure rise due to combustion for this pressure-time card is plotted in Fig. 15. The slope of this curve is almost linear and approximately one-half that of curve *A* in Fig. 10. The highest rate of pressure rise occurs 10 to 20 deg after top dead center. At this time, the piston is moving down at a fairly rapid rate.

Unfortunately, photographs of combustion in this engine are not available.

The curves plotted in Fig. 16 compare the per cent pressure rise due to combustion in the diesel engine (curves *A* in Figs. 10, 13, 15) with that of a spark-ignition engine in normal operation without detonation. The curves *D* and *C* clearly show the difference between the combustion characteristics of the two types of engines.

Maximum rate of burning must occur in the first one-third of the combustion period for satisfactory detonation control and optimum smoothness of operation in the spark-ignition engine. The data now available indicate that for best diesel performance the maximum rate of burning should occur in the last one-half of the combustion period, also that the position of top center in relation to the maximum rate of burning is of prime importance.

In conclusion, although considerable progress has been made in the standardization of the ignition quality of diesel fuels, the demand for smoother, cleaner, diesel-engine operation is increasing. Evidently combustion control will become as important a factor in diesel-engine design as it is now in the spark-ignition engine.

The recording of accurate indicator cards and the accurate determination of the rate of fuel injection under engine operating conditions is essential to this work. The lack of satisfactory instruments has been a serious handicap in the past. It is hoped that more accurate instruments will now be available as a result of development during the war.

DISCUSSION

Questions Efficiency Obtained by Author

—H. M. WILES

Waukesha Motor Co.

IT has been known and practiced by many diesel-engine builders to select the best air swirl and injection rate for a given range of operating speeds to obtain optimum performance at rated speed. The problem usually becomes a compromise between the best rated speed setting and a permissible rate of pressure rise at the lower speeds. With turbulence induced by piston speed, this feature helps keep the rate of flame travel in step with the rpm.

Mr. Bryan has shown in the curves (Fig. 16) a rate of pressure rise with an experimental diesel engine similar to that of a gasoline engine, and I am wondering if this would be possible in a commercial production engine without sacrificing some efficiency, power, and early smoke in the load range.

In the engine combustion pictures the author speaks of the flaming mass stopping revolving at 10 to 12 deg ATDC, which is not quite clear to me in studying the pictures, so possibly he can explain this detail a little further. Also, the position of the start of ignition affecting the formation of smoke is very interesting and might be enlarged upon.

In the bomb pictures (Fig. 7) showing the ignition starting at 1¾ in. from the nozzle tip, and with the running engine pictures (Figs. 8 and 11) all showing the ignition occurring at relatively the same position, it would be interesting to know what design features are most effective in controlling this ignition position.

The statement of the maximum rate of burning occurring in the last one-half of the diesel cycle is something new and probably could be further explained.

I believe we can all agree with the author's conclusion that we must design to meet the demand for smoother, cleaner diesel operation, and that combustion control is becoming a major factor, particularly since the higher rpm demand is increasing.

Mr. Bryan shows an efficient combustion-chamber design utilizing a directional throat for producing high compression turbulence in the prechamber and combustion turbulence in the main chamber.

At Waukesha, Mr. Horning used to say that an automotive engine could not run without turbulence and turbulence control, therefore, it has been a guiding principle in the design of our carburetor type of engines, our Hesselman spark diesel engines, and the compression diesel engines.

Author's Closure To Discussion

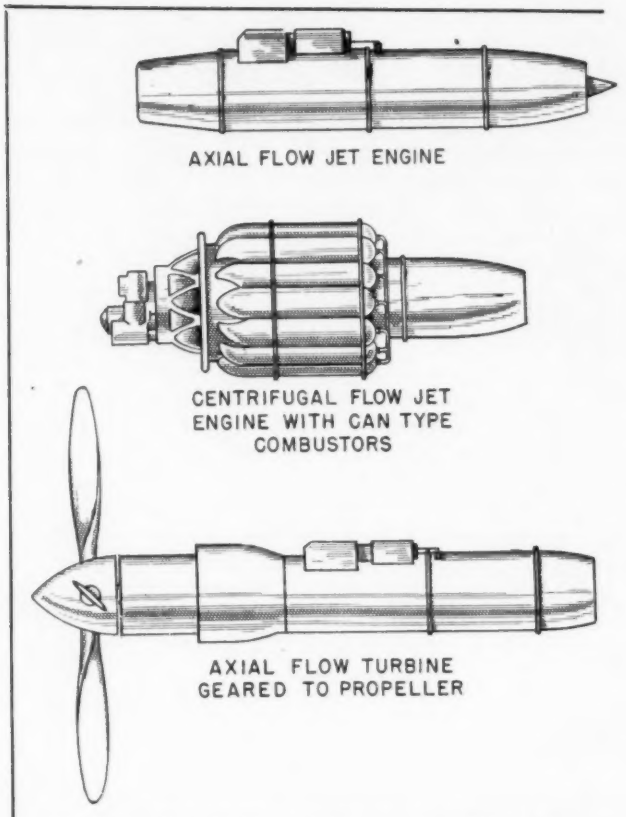
MR. WILES questions the efficiency of the engine operating under the conditions which produced the pressure-time curve shown in Fig. 14. When this card was taken, the experimental engine was producing about 10 psi higher brake mean effective pressure than the engine from which pressure-time card shown in Fig. 9 was taken, and the brake thermal efficiency was 2% higher.

The action of the revolving mass of flaming gases in the precombustion chamber is as follows. First: The mass of air in the precombustion chamber is rotating at a high rate before fuel is injected. Second: Injection, ignition, and combustion of the fuel retard this rotation somewhat, but the entire flaming mass continues to revolve. Third: Expansion caused by combustion in the precombustion chamber forces hot unburned gases into the main chamber; these mix with the fresh air in this chamber, ignite, burn, and quickly reach the precombustion chamber pressure. Fourth: As the piston velocity downward increases, the pressure in the main chamber drops, creating a pressure difference between the precombustion chamber and main chamber. Due to this difference, more hot unburned gases are discharged into the main chamber. This continues until all of the fuel is burned. When the rate of gas flow out of the precombustion chamber reaches an appreciable value the rotation of the mass of gas in it ceases.

Many factors enter into the control of the position of the ignition point in the precombustion chamber. One of the advantages of this type of chamber is the quite accurate control of turbulence, wall temperature, and mixture ratio that can be obtained by changing the relative volume, wall thickness, throat angle, and throat area.

Referring now to the question of having the maximum rate of burning in the latter one-half of the combustion period, a study of indicator cards from various types of diesel engines shows that the combustion process in these engines varies from combustion at constant volume to very near constant-pressure combustion. The difference in the rate of burning in these two processes is not readily observed until the pressure-time curves are broken down to show the pressure rise due to combustion alone. We then find that maximum rate of burning occurs in the early part of constant-volume combustion but occurs in the latter part of the constant-pressure process. Therefore, as the combustion in an engine approaches the constant-pressure process, the maximum rate of burning will occur later in the period. In our opinion, smoother operation can be obtained with less smoke when the maximum rate of burning occurs in the latter part of the combustion period. However, the rate of burning must be held within certain limits in any case.

GAS T



■ Fig. 1 – General types of aircraft gas turbine engines

IN part, this paper is a restatement of material contained in a previous paper "The Place of the Gas Turbine in Aviation" written with C. D. Flagle, Westinghouse Aviation Gas Turbine Division, as coauthor. Additional basic material contained in the present paper discusses the potentialities of gas turbine operation at low peak cycle temperatures. Recent significant improvements in gas turbine components at the Aviation Gas Turbine Division have yielded unusually high jet engine efficiency under partial thrust operation; and in light of these developments, the relationship between cycle temperature and overall jet engine efficiency is reviewed.

The gas turbine and the airplane have become firmly linked together recently, not only in the minds of newspaper and magazine readers but also in the thinking of responsible engineers and aircraft designers. The fundamentals of gas turbine engines are well known to aircraft engineers, but some of the consequences of these principles are as yet unrecognized by many individuals. It is our purpose to review here the general factors influencing the use of turbine-jet engines and turbines geared to propellers, and to indicate the effects of these factors upon the performance characteristics of airplanes.

There are two designs of gas turbines under development for aircraft use. One type uses a centrifugal compressor with either single- or double-entry air intake, and the other type utilizes an axial-flow compressor. Both use reaction turbines with single or multiple stages; combustor designs vary from the can type of the original Whittle engine to the in-line combustion chamber, which is so well adapted to the axial-flow compressor. Outline sketches of several typical engines are shown in Fig. 1, and it should be noted that the gas turbine can be utilized as a pure jet engine, or some of the output can be obtained on a shaft and geared to a conventional propeller.

Immediate developments are limited to open cycle turbines without the use of heat-recovery devices, interstage reheating in the turbine section, or intercooling in the compressor. At this time the additional weight and complication of such auxiliary equipment more than offsets the advantage of decreased fuel consumption for aircraft use. There can be but little doubt that when very large airplanes are in common use employing gas turbines of over 10,000-hp rating, waste heat recovery will become desirable, particularly in lighter-than-air craft; but for the present it does not appear to be feasible. Any tampering with the simple open cycle engine at the present time is for the purpose of gaining thrust augmentation for take-off or emergency conditions, just as water injection is used with reciprocating engines.

All air breathing engines, as distinguished from rockets which carry their own oxidizing agent as well as their fuel, compress air, add heat in a combustion chamber, and expand the hot gases to obtain useful work, either by pushing against pistons or turbine blades or by utilizing the pure jet reaction of the exhaust. In a gas turbine the division of energy which is allowed to remain in the exhaust jet and that which is delivered to a rotating shaft is under the control of the designer. The simplest engine is the pure jet engine, and it is also the most efficient, provided the ratio of the aircraft's flight speed to the exhaust jet velocity is sufficiently high to yield a satisfactory jet propulsive efficiency. When propulsion is obtained by taking in a fluid and accelerating it to a relatively high exhaust velocity, the resultant thrust is proportional to the change in velocity; thus maximum thrust with constant airflow through the engine would be obtained at standstill and there would be no thrust when the plane's speed reaches that of the jet exhaust. Actually, the ram resulting from flight speed allows the turbine to induct more air with increasing flight speed, and the thrust may increase through a portion of the speed range. This is not the case with rockets where the entire propulsive charge is carried in the rocket, and the ejection of a part of the charge from the rocket always results in an added increment of thrust regardless of the speed of the rocket.

Since the jet engine's exhaust must always leave a wake in order to obtain thrust, energy left in the wake represents a loss which is reflected in decreased jet propulsive efficiency. The higher the ratio of flight speed to the jet velocity, the lower is the wake velocity and the better the propulsive efficiency. Fig. 2 shows the jet propulsive efficiency of a typical jet engine operating at a turbine inlet temperature of 1300 F for different flight speeds; and for purposes of comparison, the efficiency curve of a variable pitch propeller is also shown. Because the velocity ratios are unfavorable below approximately 500 mph, the jet

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GAS TURBINES AND AIRCRAFT

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propulsive efficiency in this example is not as good as that of the propeller; the propeller loses out at higher speeds because of excessive tip or shank losses. In order to obtain 67% propulsive efficiency the ratio of flight speed to jet velocity must be 0.5, and in order to reach 80% propulsive efficiency the velocity ratio must be 0.67. The jet propulsive efficiency must be multiplied by the cycle or engine efficiency of the gas turbine in order to obtain gross efficiency, and the flight drag losses must also be subtracted in order to obtain net efficiency.

The compressor of the jet engine inducts air and compresses it to several atmospheres, usually to a pressure ratio in the range of 4 to 6. Higher pressure ratios can be used, but with present compressor and turbine efficiencies and with present limits on turbine inlet temperatures, very little increase in efficiency of the engine occurs if the ratio is higher. In order to illustrate this point, the familiar family of curves showing cycle efficiency against compression ratio for various turbine inlet temperatures is shown in Fig. 3. Here the compressor efficiency and turbine efficiency have each been assumed to be approximately 85% with 70 F air at the compressor intake. The same curves show the importance of operating at maximum turbine inlet temperatures if high cycle efficiency is to be obtained However, as both compressor and turbine efficiencies are further improved, the necessity for going to excessively high operating temperatures will be minimized and reasonably good cycle efficiencies will be obtained at temperatures in the 1000-1300 F temperature range. This is particularly true if the temperature of the compressor inlet air is low, as normally encountered at high altitudes. Hot air induction at the compressor is costly in terms of both cycle efficiency and output in the open cycle gas turbine, and jet engines for aircraft use must be selected with sufficient reserve thrust for satisfactory take-off performance on hot days. Auxiliary means of thrust augmentation may be necessary under such conditions if the engines are selected solely on the basis of high-altitude, low-temperature performance; additional thrust can be obtained with rockets or by special provisions for burning more fuel within the jet engines without exceeding permissible temperature limits in the turbine.

In addition to the effects of temperature upon turbine operation, the mass of air inducted, of course, decreases at higher altitudes, and since the thrust depends upon total momentum change, thrust also decreases with increasing altitude. Fortunately, the lower temperatures encountered at higher altitudes partially compensate for the decrease in mass flow through improvement in efficiency, and the output of either a jet engine or a geared gas turbine does not fall off as rapidly with increasing altitude as does that of the unsupercharged reciprocating engine, provided compressor efficiency is not adversely affected by low-temperature operation. Fig. 4 shows typical power availability curves for both reciprocating engines and gas turbines at different altitudes.

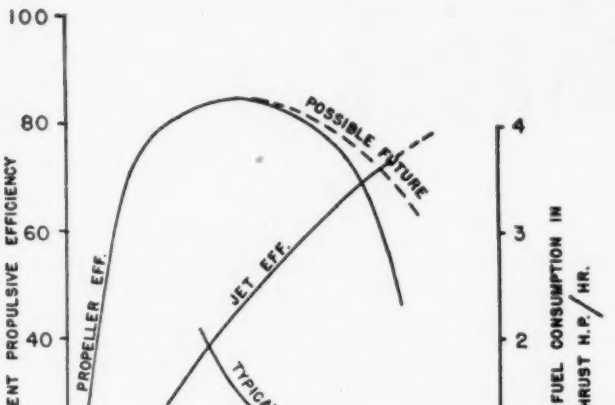
When the output of the gas turbine is used to drive a propeller, the thermal efficiency of the turbine is relatively independent of flight speed and the net efficiency of the turbine-propeller combination will be good through a wide range of aircraft speeds. With the variable pitch propeller, maximum thrust is obtained at take-off; at high flight speeds the propeller is in higher pitch and the same shaft power is available with lower propeller thrusts. The jet engine, on the contrary, does not have any provisions for "shifting gears" by means of a variable pitch propeller, and with high turbine temperatures the thrust will be relatively constant from take-off to maximum speeds. Fig. 5 shows relative thrusts of a geared turbine driving a propeller and of a jet engine with the same power ratings at 375 mph. The propeller turbine has about four times the take-off thrust of the jet engine, but will begin to lose out rapidly in thrust and power availability at very high flight speeds.

Both the jet engine and the geared turbine operate at relatively constant rpm when compared with the reciprocating engine. The compression ratios of both centrifugal- and axial-flow compressors decrease very rapidly with decreases in rpm; and since present turbine designs operate on the low side of the optimum compression ratio range, a small drop in compressor speed will result in a decrease in thrust. For maximum efficiency operation, aircraft gas turbines are essentially constant rpm engines, with but minor changes in speed desirable at various altitudes and for different load conditions. The use of propellers in conjunction with gas turbines thus necessitates a new philoso-

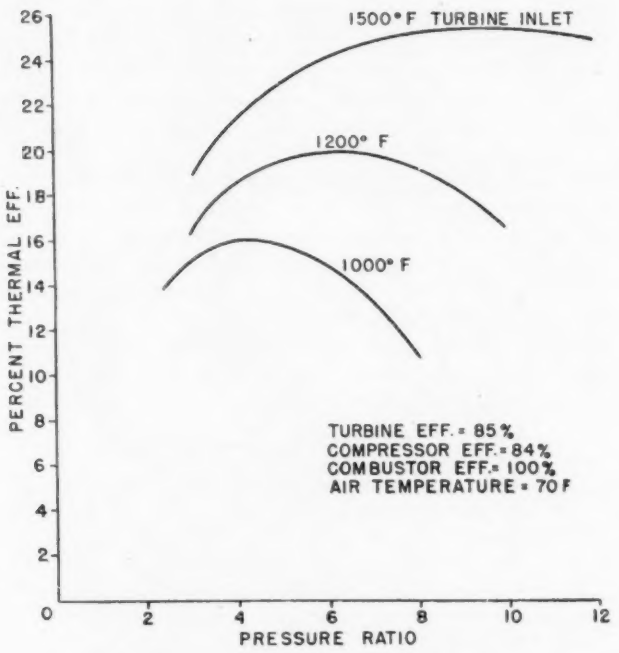
RECENT significant improvements in gas turbine components have yielded unusually high jet engine efficiency under partial thrust operation, the author reports, and he reviews the relationship between cycle temperature and overall jet engine efficiency, in the light of these developments.

He considers the general factors influencing the use of turbine-jet engines and turbines geared to propellers, and indicates the effects of these factors upon the performance characteristics of airplanes.

THE AUTHOR: F. W. GODSEY, JR. (M '45), manager of the New Products Division, Westinghouse Electric Corp., Pittsburgh, joined Westinghouse in 1940 as an advisory engineer. Just prior to his association with Westinghouse, he was affiliated with Sprague Specialties. He was graduated from Rice Institute in 1927 with the degree of B.S. in E.E. and from Yale University in 1929, where he received his M.S. in E.E.



■ Fig. 2 – Propulsive efficiencies of propellers and jets of jet engines (fuel consumption of typical jet engine also shown)

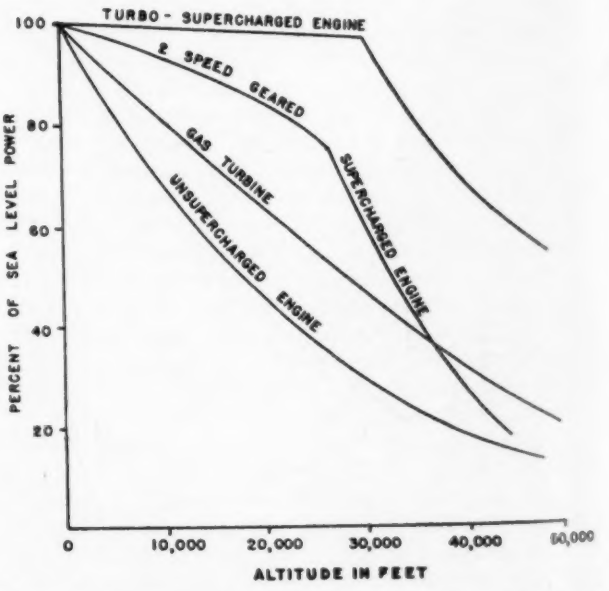


■ Fig. 3 – Thermal efficiency of open cycle gas turbine

phy of propeller control in flight, and may result in major design changes in propellers. It is probable that propeller change gears for two-speed operation will be desirable for extremely high speed aircraft with large powerplants in the future.

The jet velocity of the jet engine depends upon thermal efficiency of the engine and the turbine temperature; there are other factors of importance, such as ram due to flight speed and the ambient temperature of the atmosphere, but the major influence on jet velocity with a given engine is the turbine inlet temperature. Since the jet propulsive efficiency is directly connected with the ratio of flight speed to jet velocity, for any flight speed there is an optimum turbine operating temperature for best overall efficiency. This is a point that is not well understood by many engineers and airplane designers; higher temperature results in more thrust, but it also results in a decrease in jet propulsive efficiency at the same time. If the jet propulsive efficiency decreases with increasing temperature faster than the cycle efficiency increases at the same time, there will result a net loss in efficiency at a given flight speed.

Until recently, compressor and turbine efficiencies have been so low that gas turbine thermal efficiencies were impractical for aircraft use with open cycle turbines at any but extreme turbine temperatures. In the last few years significant improvements have been made in the efficiencies of gas turbine components, so that good cycle efficiencies can be obtained in the 1200 F range, only slightly above the temperatures now used in some large steam turbines. However, even at 1250 F, the high-altitude jet velocity is of the order of 1500 fps, so that flight speeds of 550 mph must be maintained in order to obtain jet propulsive efficiencies of 70%. Jet engine components have been improved to the point where partial load operations can be carried out relatively efficiently at low turbine temperatures, and the resulting jet velocities are sufficiently low so that fairly good energy conversion is obtained at moderate flight speeds. Fig. 6 shows the calculated specific fuel rates obtainable from a jet engine at different turbine temperatures, and relative thrusts are also shown for two different ambient temperatures. Fig. 7 shows the curves for thrust power fuel rates of the same engine at different temperatures and different flight speeds, fuel rates are shown in pounds of fuel per thrust horsepower-hour and include an allowance for engine nacelle drag, usually negligible except at low thrust conditions in the case of the small diameter axial-flow engine with straight-through



■ Fig. 4 – Effect of altitude operations on power availability of various typical aircraft engines

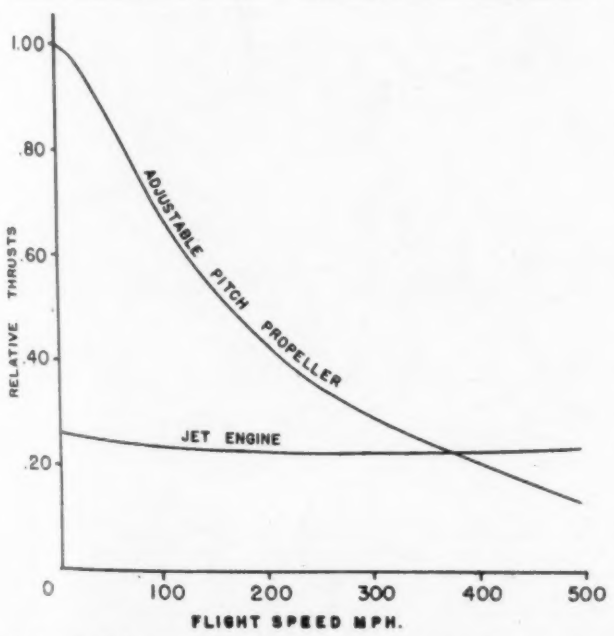
combustion chamber. The example chosen for Fig. 7 is based on a ram pressure recovery of roughly 70%. Some experimental jet engine installations have exhibited considerably better ram recovery than chosen here. Increases in this pressure recovery are quite beneficial to fuel rates and available thrusts.

Particular attention is called to the fact that high turbine inlet temperatures do not always result in best efficiency. It is apparent from Fig. 7 that turbine temperatures of the order of 1500 F may result in lower efficiency operations at flight speeds under 500 mph than when turbine temperatures are in the 1000-1300 F range. Further improvements in turbines and compressors will not materially change this situation, but they will make it possible to achieve greatly reduced fuel rates at low temperatures and also at low flight speeds.

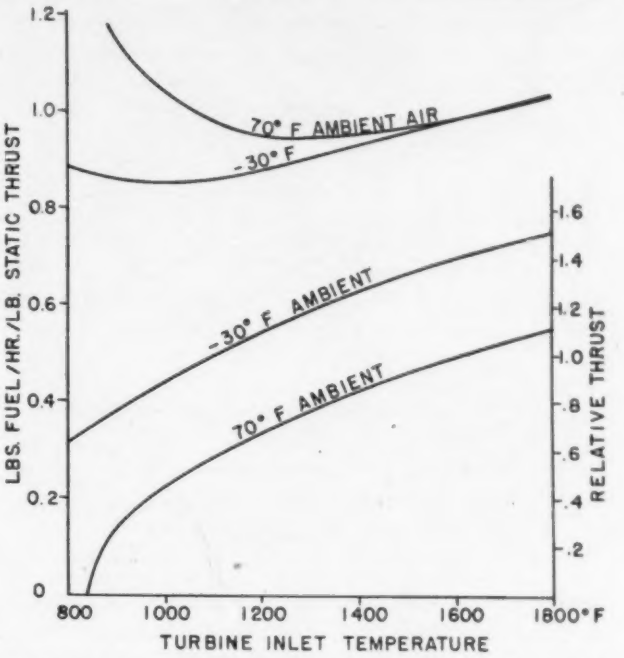
It is fundamental of the simple jet engine that higher than present operating temperatures with consequent higher jet velocities cannot be used to increase flight range until supersonic flight speeds are required. Higher temperatures will make possible greater thrusts for take-off and emergency operations and will reduce the specific weights of jet engines, but they are not desirable for greater flight economy at any cruising speeds now contemplated.

Thus, for the jet engine, available high-temperature alloys capable of use at temperatures to 1500 F are probably adequate for our present turbine blading needs except for take-off conditions. The most fruitful fields of research in jet engine design are compressor, combustor, and turbine efficiency improvements.

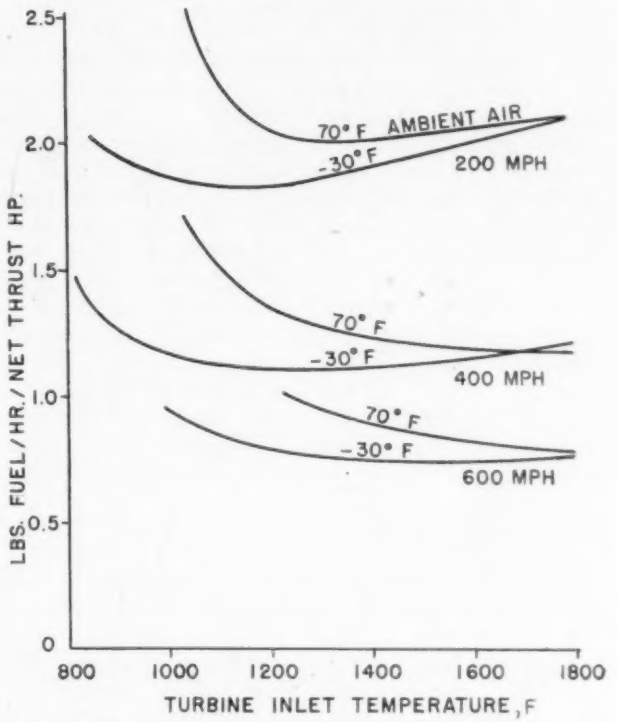
The foregoing comments do not apply to the gas turbine geared to a propeller. The maximum possible energy conversion is usually desirable on the shaft of the turbine, and the jet velocity is reduced to the point where only a small portion of the remaining available energy is represented as leaving energy of the turbine exhaust. Therefore, the propulsive efficiency in flight depends upon propeller characteristics instead of the jet velocity, and the overall efficiency of the powerplant depends principally upon the thermal or cycle efficiency of the engine in the range of flight speeds where good propeller efficiency is obtained. It may be said that a desirable upper limit on geared turbine inlet temperatures is established only by turbine and compressor efficiencies and practical compres-



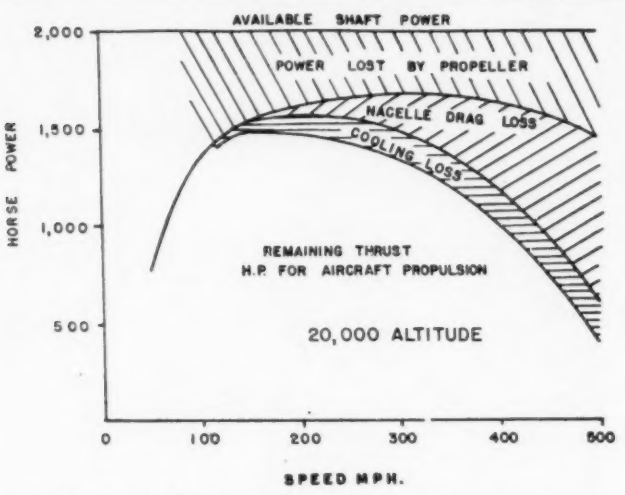
■ Fig. 5 – Relative propeller and jet engine thrust with equal thrust power at 375 mph, at different flight speeds



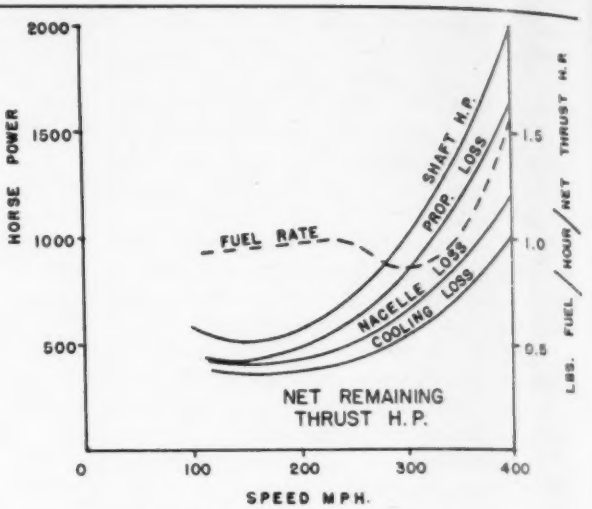
■ Fig. 6 – Effect of turbine temperature upon static thrusts and fuel rates of jet engines



■ Fig. 7 – Effect of flight speed and turbine temperatures on jet engine efficiency



■ Fig. 8 – Typical losses associated with large reciprocating engine in wing nacelle and operated at full power



■ Fig. 9 – Typical flight losses of powerplant of Fig. 8 when operated on power-required curve of large airplane designed for long-range operations

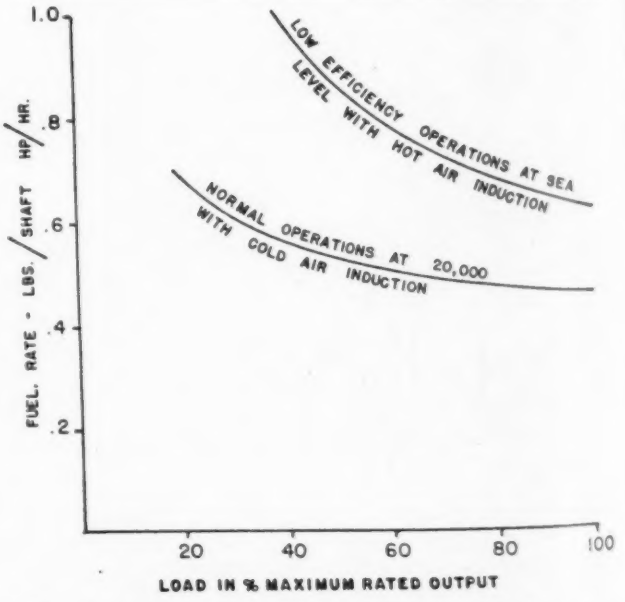
sion ratios. Therefore, it is necessary and desirable to continue research in the metallurgy of high-temperature alloys for the extension of geared turbine temperatures to still higher ranges.

The jet engine fuel rates per net thrust horsepower-hour which appear in Fig. 7 could not be compared favorably with the rates customarily thought of as applying to reciprocating engines, where fuel rates of less than 0.5 lb per shaft hp per hr are usual at cruising conditions. However, when the reciprocating engine is burdened with propeller losses, cooling losses, and nacelle drag, the situation begins to change. Fig. 8 shows the losses for a reciprocating engine installation in a large airplane under maximum power condition; at high speeds there isn't much thrust power left to pull the airframe through the air. Fig. 9 shows the performance of the same powerplant along the power-required curve of a moderately fast airplane. In the speed range of 250 to 325 mph the engine is operating under cruising conditions at best engine efficiency, but propeller, cooling, and nacelle losses push the fuel rates up until they begin to approach those of the jet engine. Of course, an airplane designed to cruise at 200 mph simply cannot yet carry enough fuel to use jet engines for any practical purpose, but cruising at over 400 mph definitely indicates jet engines.

High-speed flight is still expensive in terms of power requirements, even with laminar-flow wings and clean design, and it is in this region that the jet engine surpasses the reciprocating engine and propeller. Further improvements in the efficiencies of gas turbine components will result in good cycle efficiencies at lower turbine temperatures, and with lower jet velocities, and it is expected that jet engines will become competitive with reciprocating engines at much lower flight speeds in the future.

The use of a gas turbine to drive a propeller through a reduction gear results in good propulsive efficiency at both low and high speeds, at the expense of the added weight and complexity of the gear and propeller. In addition, the geared gas turbine requires higher compression ratios and higher operating temperatures than are desirable for jet engines. These disadvantages are more than offset by the high take-off thrusts available with variable pitch propellers. A jet engine with 3000 lb of maximum thrust at speeds around 400 mph would have 3000 lb, or slightly more, of thrust at take-off; a geared gas turbine driving a propeller with a total thrust of 3000 lb available at 400 mph would be able to deliver about 10,000 lb of thrust at take-off.

In order to hold the physical dimensions of the turbine in conformity with the compressor and combustor dimensions and to avoid gearing between compressor and turbine, the turbine disc area is small and gases leave the last row of turbine blades with a fairly high residual velocity. This leaving loss may be of the order of 15 to 25% of the available energy at the propeller shaft; of course, if the designer wishes to, a much smaller amount of power can be removed on the shaft and a large jet thrust may be available. Ordinarily though, the turbine exhaust would leave at a velocity that would result in good propulsive



■ Fig. 10 – Range of fuel consumption rates for aircraft gas turbines with reduction gear output to propeller shaft (rates not adjusted for exhaust jet thrust effects)

efficiency in the 400-mph flight speed range and perhaps 15% of the total thrust of the powerplant would be due to the jet effect of the exhaust at 400 mph.

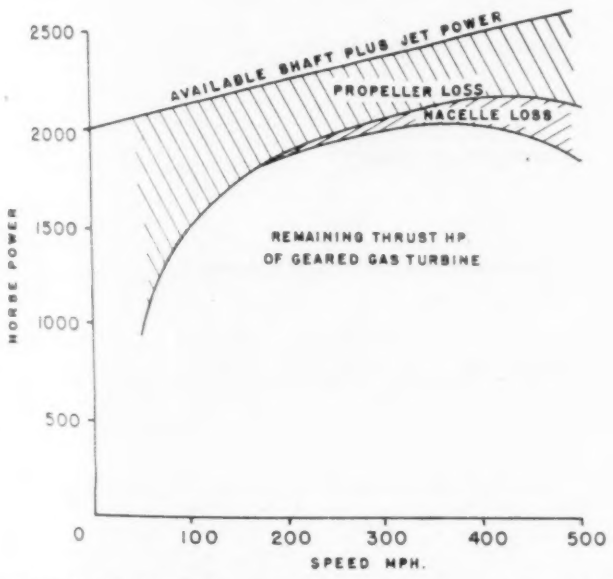
Under these conditions, the geared gas turbine is a reasonably efficient powerplant that cannot quite compete with the best teststand fuel rates of the reciprocating engine, but can show excellent economy of operation in the airplane because of extremely low nacelle drag, a practical absence of cooling losses, and minimum fuel rates at maximum output. Fig. 10 shows typical ranges of fuel rates for a geared turbine. Shaft efficiency is good in a well designed gas turbine if operating conditions are favorable, that is, if the ambient air temperature is low and the turbine is operated at over 50% of its rating. When air temperatures are high, in the vicinity of 100 F, both efficiency and power output drop rapidly. With the turbine inlet temperature at 1200 F changing compressor inlet air temperature from 50 F to 120 F will result in almost a 30% drop in output with an increase in fuel rate of over 20%; however, if the inlet air temperature is decreased the fuel rate decreases approximately 3% for every 10 F drop in temperature and the power availability increases by about 4%. For a normal flight plan at reasonably high altitudes, low efficiency operation is limited to take-off and climb conditions. These operating characteristics of geared gas turbines also apply to jet engines.

The axial-flow gas turbine with in-line combustor presents a very small frontal area, less than one-fourth that of the equivalent reciprocating engine. The centrifugal-flow gas turbine and the axial-flow turbines with can type of combustors are not as small in diameter, but they also have a much lower parasitic drag than does the reciprocating engine powerplant. Fig. 11 shows typical flight losses and maximum power availability for a 2000-hp turbine driving a propeller. The shaft power available is almost constant with speed and the exhaust jet power increases with speed. The increment of jet power is almost enough to offset propeller and nacelle drag losses in this example throughout the useful flight speed range of present aircraft. Close to the rated shaft output of the gas turbine can be had as thrust horsepower at speeds up to and beyond 500 mph; Fig. 11 should be compared with Fig. 8 for perspective.

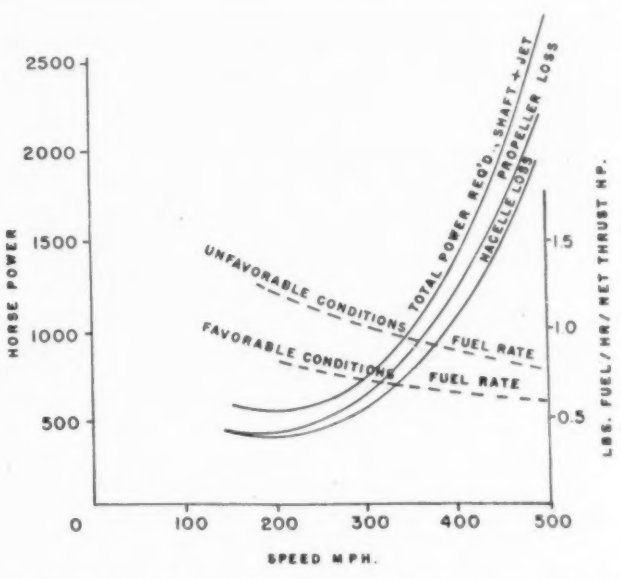
When the gas turbine of Fig. 11 is used in a high-speed airplane designed for gas turbine powerplants, performance of the order of that shown in Fig. 12 will be obtained. Nacelle drag and propeller losses are shown added to the airframe power-required curve in order to obtain net thrust horsepower requirements for the propeller turbine powerplant. Fuel rates for the net thrust horsepower of the powerplant are shown; and it is evident that partial-load operation of the gas turbine and warm air induction both result in much increased fuel rates. However, it is equally evident from a comparison of Figs. 12 and 9 that only under the most unusual and unfavorable operating conditions will the geared gas turbine fail to deliver more thrust energy to the airframe per pound of fuel than can be obtained from conventional powerplants.

The performance characteristics of aircraft using either jet engines or turbines geared to propellers are readily approximated from the known characteristics of the two types of powerplants, and can be compared with reciprocating engine powered aircraft.

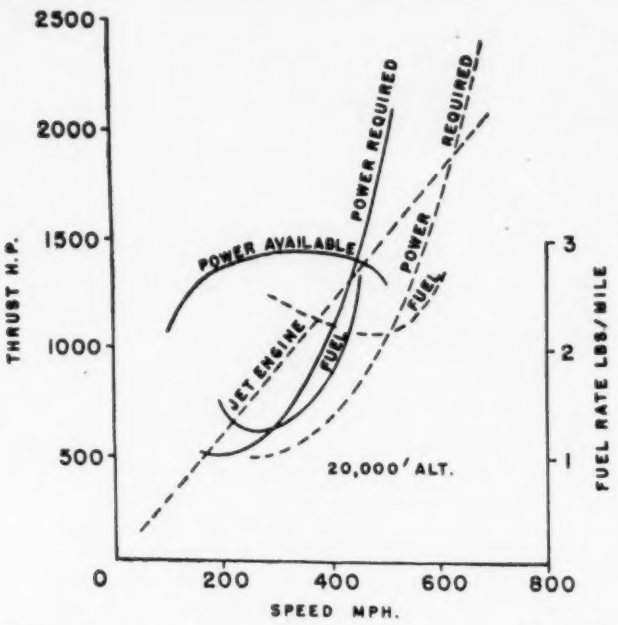
The most suitable applications for jet propulsion appear to lie in speed ranges where compressibility effects are the principal unknowns, although improvements in jet engines are now dropping the useful lower speed ranges at a satisfactory rate. Projecting the estimates somewhat into the future, a jet fighter can be compared with the present best reciprocating engine fighter; and to give the jet plane every possible advantage, its drag is assumed to be half that of the conventional plane due to refinements made possible by the use of the jet engine. Flight conditions at 20,000-ft altitude are shown in Fig. 13, solid for the conventional aircraft and dotted for the jet fighter. Even with the reduction in drag of the jet plane, fuel used per mile of



■ Fig. 11 – Typical flight losses estimated for gas turbine engine with geared propeller (Note that jet thrust almost offsets propeller and nacelle losses through large part of speed range shown)



■ Fig. 12 – Typical flight losses of geared gas turbine powerplant of Fig. 11 when operated on power-required curve of high-speed aircraft at 20,000 ft altitude



■ Fig. 13 – Power required and power available curves for single-engine airplane with reciprocating engine (solid lines) and jet engine (dotted lines) (fuel consumption rates for both aircraft shown)

travel of the jet plane is about twice that of the conventional aircraft. However, the reciprocating engine aircraft flying at a high cruising speed of 270 mph for maximum range would be flying only about half as fast as would the cruising jet fighter at 500 mph.

With auxiliary fuel tanks and with extra fuel possible because of the low specific weight of the jet engine, the jet fighter could climb to 30,000-ft altitude for better cruising conditions with only 40% more fuel per mile than the conventional aircraft and could stretch its extreme range to something in excess of 2000 miles. The low take-off thrust of the jet engine would not permit the aircraft to get off of the ground with as heavy a fuel overload as can be accomplished with the reciprocating engine and propeller, so that the extreme range of the jet fighter is still inferior to that of the conventional aircraft.

The bar chart of Fig. 14 shows some rough comparisons between the jet fighter and the reciprocating engine fighter. While the values may be shifted somewhat by changing relative powerplant ratings, in general the jet fighter excels only in speed and high-altitude performance. Improvements in jet engines should eventually result in superior characteristics in other respects.

The fighter plane of Figs. 13 and 14 could be equipped with a gas turbine-propeller powerplant instead of either the jet engine or the reciprocating engine. However, detailed comparisons of these three types of fighter aircraft are similar to comparisons that can be drawn for much larger aircraft of the heavy bomber or transport class, and an example will be chosen in that class.

The gas turbine-propeller powerplant is equally adaptable to fighters or large airplanes. The improvement in fuel efficiency with increasing air speed and power results in somewhat higher cruising speeds than for aircraft with reciprocating engine powerplants. The decrease in available power at higher altitude makes it necessary to install

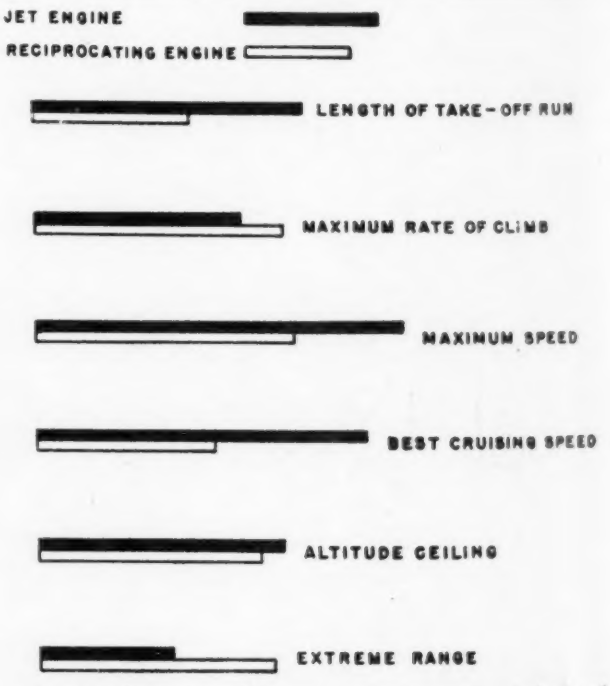
Table 1 – Take-Off Conditions

	Gross Weight, lb	Installed Powerplant Weight, lb	Fuel Weight, lb	Take-Off Thrust, lb
Conventional	120,000	20,800	40,000	28,000
Turbine-Propeller	120,000	15,200	45,800	44,000
Jet Propelled	120,000	8,000	52,800	24,000

gas turbine powerplants that may have considerably more power at sea level and for low-altitude climb than is now considered normal in airplane design. This is an advantage rather than a disadvantage however; a propeller able to absorb the turbine's shaft output at 20,000 or 30,000 ft can efficiently handle the greatly increased output of the same turbine at sea level, due to the denser air at low altitudes. Thus, a turbine powerplant with a rating equal to that of a supercharged conventional engine at 20,000-ft altitude might have a take-off thrust at sea level 60% in excess of the thrust of the conventional engine. A much shorter take-off run than normal is, therefore, possible, and rate of climb is greatly superior to the conventional aircraft. The field of usefulness of the gas turbine with propeller extends to all aircraft requiring engines of greater than perhaps 1000-2000 hp, and designed for all speeds up to and in excess of 500 mph.

The total thrust horsepower requirements of a typical large (4-engine) airplane at 20,000-ft altitude are shown in Fig. 15. Curve *A* is for the plane with reciprocating engines and includes all powerplant drag losses but does not include propeller losses. Curve *B* is for the same airplane but with gas turbine engines geared to propellers. Curve *C* is for a jet propelled aircraft similar in appearance to the turbine-propeller plane but without the propellers. The gross weight in each instance is assumed to be 120,000 lb and powerplant plus fuel weights are held constant.

The conventional engine installation is four engines supercharged to 2000 hp each at 20,000 ft. In the jet pro-



■ Fig. 14 – General performance comparisons for airplanes of Fig. 13

Table 2 – Cruise Conditions (20,000-Ft Altitude)

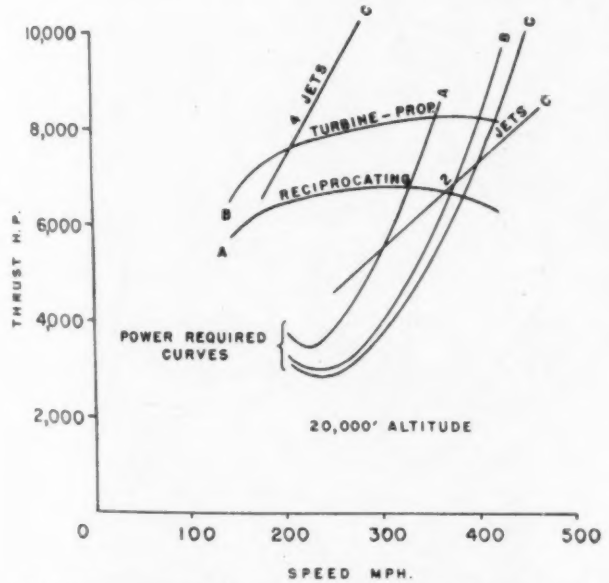
	Best Cruising Speed, mph	Miles per Lb Fuel	Extreme Range, miles
Conventional	250	0.120	4800
Turbine-Propeller (4 engines)	300	0.125	5700
Turbine-Propeller (2 engines)	280	0.145	6600
Jet Propelled (2 engines)	350	0.050	2650
Jet Propelled (4 engines) (35,000-ft altitude)	460	0.066	3500

pelled version, four 6000-lb static thrust engines are used; this will give 24,000-lb take-off thrust, approximately equal to the take-off thrust of the plane with reciprocating engines. In the gas turbine geared propeller version, four engines capable of delivering 2000 shaft hp each at 20,000 ft are used. At take-off these engines will deliver 3200 hp each. Maximum power availability curves for these powerplants are also shown in Fig. 15, power availability is shown for flight with either two or four jet engines in use.

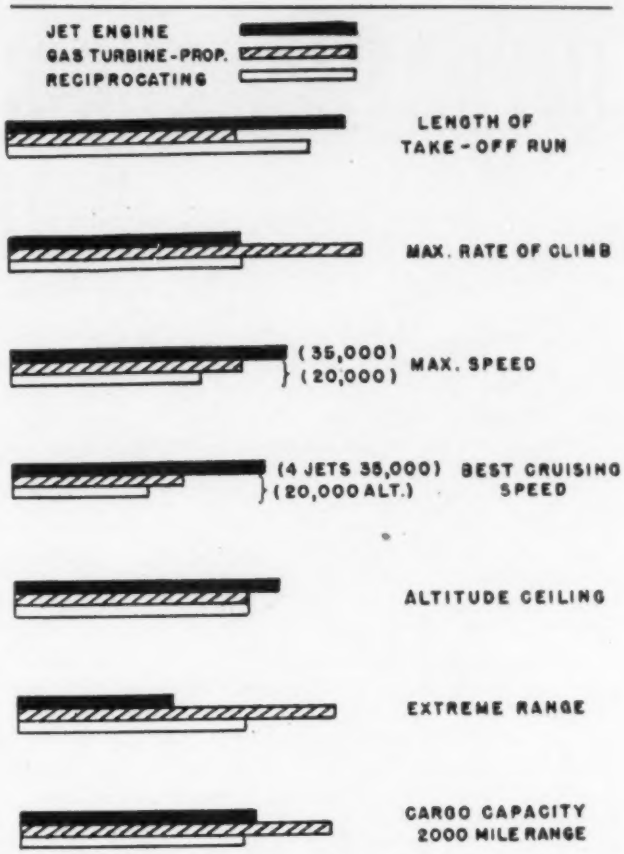
Thus, at take-off the three versions appear as in Table 1.

Cruising conditions are shown in Table 2; jet cruising is on four engines at 35,000 ft and on two engines only at 20,000 ft, since partial load operations with four engines at that altitude would result in abnormally low turbine temperatures and very poor cycle efficiency. Fuel consumption with four jet engines operating is prohibitive for any conditions other than at take-off or very high altitudes. Cruising with four gas turbine-propellers in use is tabulated, and also with two turbine-propellers in use and two feathered; a suitable drag allowance has been made for the feathered propellers.

The sensitivity of the gas turbine-propeller to partial-load operation is seen in Table 2, an 18% increase in range is obtained by operating two engines close to full load and maximum efficiency rather than operating four engines at less than half load. This is a fundamental difference that exists between turbines and reciprocating engines, which makes necessary a careful selection of turbines and reciprocating engines, and also requires a careful selection of turbine ratings for any given application if maximum fuel economy is to be gained. In the case of jet propulsion,



■ Fig. 15 – Power-available and power-required curves of 120,000-lb airplane with four engines (curves *A*: reciprocating engines, *B*: gas turbines geared to propellers, and *C*: jet engines)



■ Fig. 16 – General comparison of characteristics of airplanes of Fig. 15 and Tables 1 and 2

since high speed and high propulsive efficiency go together, very high altitude operation is essential to maximum flight economy.

The general characteristics of the three airplanes described in Tables 1 and 2 are compared in Fig. 16. It is interesting to note that the gas turbine-propeller airplane is superior to the conventional aircraft on every count with the exception of altitude ceiling, where they are about equal with reciprocating engines turbo-supercharged. Since turbine-propeller fuel economy and installed powerplant weights are both superior to those of the conventional engine, the long-range and short-range load-carrying abilities of the turbine propeller aircraft are superior to the conventional one. The jet propelled aircraft has a very light powerplant installation and can get off the ground for short flights with about five tons more payload than the conventional aircraft, and about three tons more than the turbine-propeller version. However, jet propulsion requires so much more fuel than other types of powerplants that at about 2000-miles extreme range the jet plane loses out to the reciprocating engine and is far behind the geared turbine in payload carrying ability.

As was found in the case of the jet propelled fighter, the jet bomber or cargo plane must, at present, be limited in use to relatively short-range, very high-speed, high-altitude operations which can stand the expense of high fuel costs. The gas turbine with a geared propeller, however, has immediate applications in all classes of service where requirements are much above 1000 hp at usual flight altitudes, and should be ready for use when aircraft designs require 10,000 hp or more in a single powerplant unit.

IMPROVING ENGINE PARTS by DIRECT MEASUREMENT

THE possibility of increasing the life and output of machines as the result of stress analysis has captured the interest of engineers and technicians. By careful work, remarkable progress has been made in the development of reliable stress analysis techniques and in the collection and interpretation of strain data.

The inability to estimate stress values or to predict the life of machine parts by mathematical calculations has stimulated the use of brittle lacquers, mechanical extensometers, and electrical strain gages for the direct measurement of the strains which prevail under load at the surface of an actual machine part.

Methods used for stress determination can be found in published literature.[1, 2, 3] The purpose of this article is the interpretation of some of the results of stress analysis.

The use of stresses as calculated from assumed loadings as a criterion of the strength of a design may be misleading. The inadequacy of this practice arises from many unknown factors. One of these is the fact that some level of residual or prestress prevails in a great many machine parts when assembled, and the stress resulting from the imposed load is superimposed on these residual or prestresses. These stresses may be set up intentionally or may result from the fabricating procedure and may cause a combined stress condition very difficult to calculate.

In determining the safe allowable imposed stress value, recognition must be given to the residual stress and its influence on the allowable imposed stress. The principal stresses in an operating machine and their origins may be classified as follows:

1. Imposed stresses resulting from operating loads.
2. Residual stresses arising on the release from nonuniform plastic deformation.
   a. Quenching.
   b. Machining.
   c. Nonuniform cold plastic yielding during fabrication, assembly, and operation.
3. Prestresses introduced during assembly.
4. Thermal stresses set up during operation as a result of thermal gradients.

■ Imposed Stress

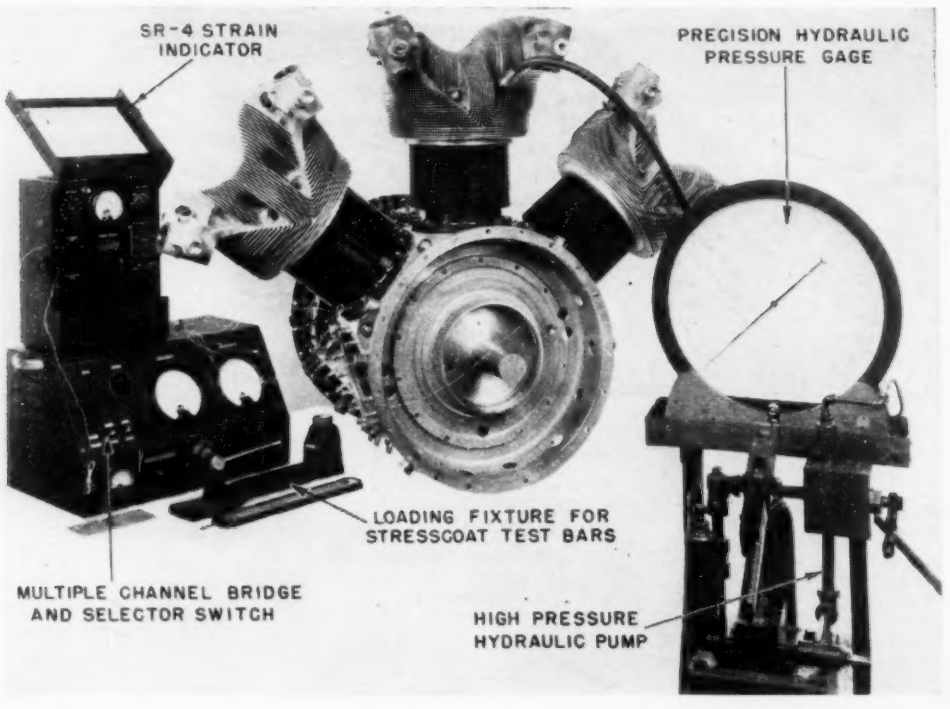
Fig. 1 shows the equipment used to load a German

[This paper was presented at the SAE National Aeronautic Meeting, New York, April 3, 1946.]

[1] See SAE Transactions, Vol. 51, April, 1943, pp. 105-124: "Methods of Stress Determination in Engine Parts," by C. Lipson.

[2] See Proceedings of Society for Experimental Stress Analysis, Vol. 1, No. 1, 1943, pp. 46-53: "Practical Strain Analysis by Use of Brittle Coatings," by G. Ellis.

[3] See ASME Transactions (Journal of Applied Mechanics), Vol. 9, December, 1942, pp. 184-188: "Brittle Coatings for Quantitative Strain Measurements," by A. V. de Forest, G. Ellis, and F. B. Stern, Jr.


SR-4 STRAIN INDICATOR
PRECISION HYDRAULIC PRESSURE GAGE
LOADING FIXTURE FOR STRESSCOAT TEST BARS
MULTIPLE CHANNEL BRIDGE AND SELECTOR SWITCH
HIGH PRESSURE HYDRAULIC PUMP

■ Fig. 1 – BMW engine crankcase set up for strain analysis using hydraulic pressure to simulate explosion pressure (stresscoat and resistance wire strain gages used for strain measurements)

OF STRAIN

by R. G. ANDERSON

Development Division
Cleveland Research and Development Laboratories
Aluminum Co. of America

BMW crankcase center section with a hydraulically simulated explosion pressure of 1500 psi in the combustion chamber of one cylinder. Fig. 2 shows the location and magnitude of the strains so developed. These strains have been evaluated by the use of stresscoat and indicate the direction and amount of strain under the specific conditions.

Additional strains are generally present as a result of unbalanced crankshaft forces. These additional strains may increase or decrease the level of the strains resulting from explosion loads. Provided no change in the direction of principal strain takes place due to these additional strains, biaxial strain measurements can be obtained from two gages at each interesting location.

After locating the higher strains and their direction by the use of stresscoat, strain gages capable of operation at elevated temperatures can be mounted at the interesting locations and dynamic strain readings obtained with the engine in operation. By mounting two strain gages at each location, one parallel and one normal to the direction of principal strain as indicated by the cracks in the stresscoat, the strains can be used to compute stress values.

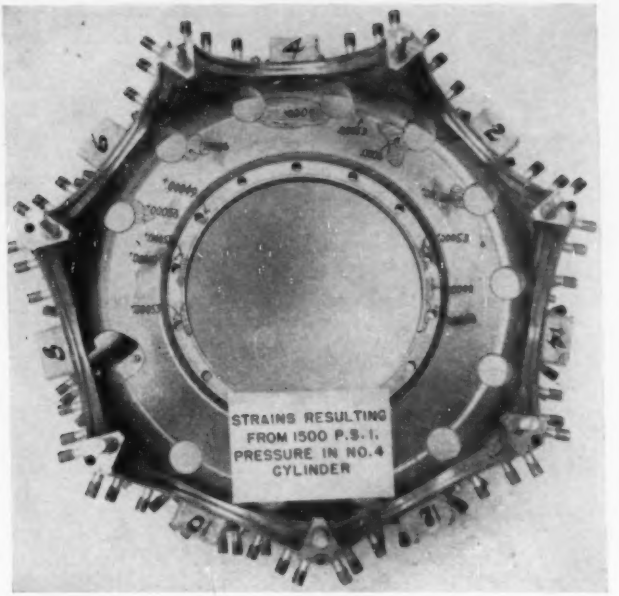
In Fig. 3 are shown formulas for computing the indicated stress resulting from biaxial strain. When the principal strain is multiplied by the modulus of elasticity of a material, the value obtained is expressed as apparent stress. A high degree of accuracy is obtained only when the principal strain is within the elastic range of the material and the strain results from pure tension or compression.

■ Residual Stress

Residual stress may be set up in a part during casting, fabrication, heat-treatment, machining, or assembly by nonuniform cold working of the material.

With the exception of the X-ray method, residual stress determination is destructive to the part. The Sachs[4] method of residual stress determination is based on the fundamental phenomenon that the removal of a part of a stressed body causes the remainder to suffer elastic strain.

Biaxial residual stress determinations at the surface of a part may be made by attaching wire strain gages at locations of interest and then carefully removing all the metal from the section except just that required for support of the gages.[5] The approximate residual strains may be deduced from the readings of the gages.



■ Fig. 2 – Stresscoat pattern outlined on inside surface of center section of BMW crankcase showing magnitude of strain

A qualitative method of residual stress determination may be made by the use of stresscoat. In Fig. 4 is shown a partial view of the head of a forged and heat-treated piston which has been coated with stresscoat. Careful drilling with a ⅛-in. sharp drill to a depth of approximately ⅛ in. caused a redistribution of stress to take place, thus causing the stresscoat to crack. Cracks in the coating concentric to the drilled holes indicate that a state of compressive residual stress prevailed at the surface of the piston head. If the cracks had radiated from the drilled hole, a state of residual tensile stress would be indicated. The random cracks in the stresscoat between the drilled holes

PRESENTED here is an interpretation of some of the results of stress analysis – including a few of the many methods whereby the life of machine parts can be improved.

Machine parts, the author states, generally contain a sufficient amount of material, but the material is often inefficiently distributed.

The author points out that in designing it is important to proportion parts so as to take advantage of beneficial prestress.

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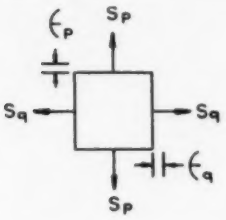
[4] See "Practical Metallurgy," by G. Sachs and K. R. Van Horn. Published by American Society for Metals, Cleveland, Ohio, 1940.

[5] See Proceedings of Society for Experimental Stress Analysis, Vol. 1, No. 1, 1943, pp. 10-18: "Residual Stresses and Fatigue Studies," by O. J. Horger, H. R. Neifert, and R. R. Regen.



$$S_p = \frac{E}{1-\mu^2}\left[\epsilon_p + \mu\epsilon_q\right]$$

$$S_q = \frac{E}{1-\mu^2}\left[\epsilon_q + \mu\epsilon_p\right]$$

$$\epsilon_p = \frac{S_p}{E} - \mu\frac{S_q}{E}$$

$$\epsilon_q = \frac{S_q}{E} - \mu\frac{S_p}{E}$$

WHERE
E = MODULUS OF ELASTICITY
μ = POISSON'S RATIO

■ Fig. 3 – Formulas for calculating indicated stress resulting from biaxial strain

result from a balanced biaxial stress. For illustrative purposes, this piston has been heat-treated in a manner which sets up a relatively high level of residual stress in the piston head.

■ Quenching Stresses

Any nonuniform temperature distribution within a part induces stresses. The distribution and magnitude of the stresses may be affected if some portions yield during cooling. Residual stresses can only be produced if portions of the part are plastically deformed during cooling. The portion which cools most rapidly retains compression stresses, whereas the part which is last to reach room temperature contains residual tensile stresses. A favorable residual stress distribution for parts subject to repeated stressing can often be obtained by quenching the part in a predetermined manner.

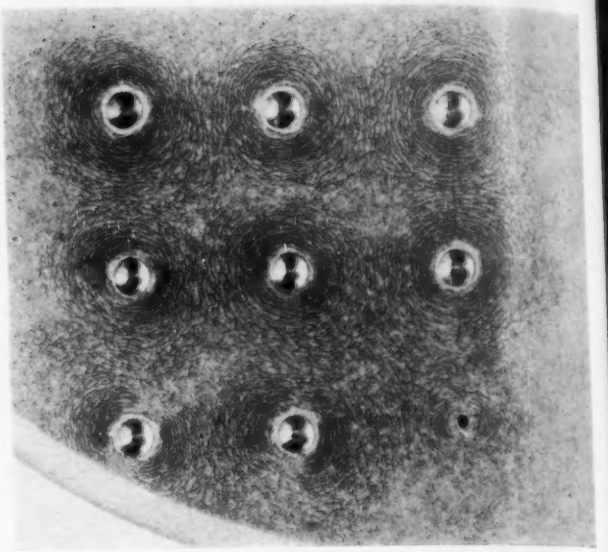
■ Machining Stresses

Stresses of this nature can be set up in the part as a result of machining operations. Surface working produced by a sharp cutting tool, where the effect is limited to a few thousandths of an inch in depth, does not materially increase the fatigue strength of a part. Surface rolling in which an appreciable amount of plastic deformation has taken place in the surface layers may considerably improve the fatigue resistance. The usual distribution of residual stresses in a body rapidly cooled from an elevated temperature is such that the stresses at the surface of the body are compressive and, in the interior, tensile in nature. Removal of a portion of the body by machining may bring about a less desirable distribution of stresses. For example, the nature of the stresses in the surface layers may change from compressive to tensile.

■ Nonuniform Cold Reduction

Residual stresses resulting from plastic yielding during fabrication, assembly, or operation may be beneficial or detrimental to the fatigue life of machine parts. This depends on the relation of the residual stress direction to the direction of the imposed stress and upon the conditions which may be set up by the different stresses. Beneficial amounts and directions of prestress can be obtained by

[6] See SAE Transactions, Vol. 51, July, 1943, pp. 248-268: "Shot Blasting to Increase Fatigue Resistance," by J. O. Almen.



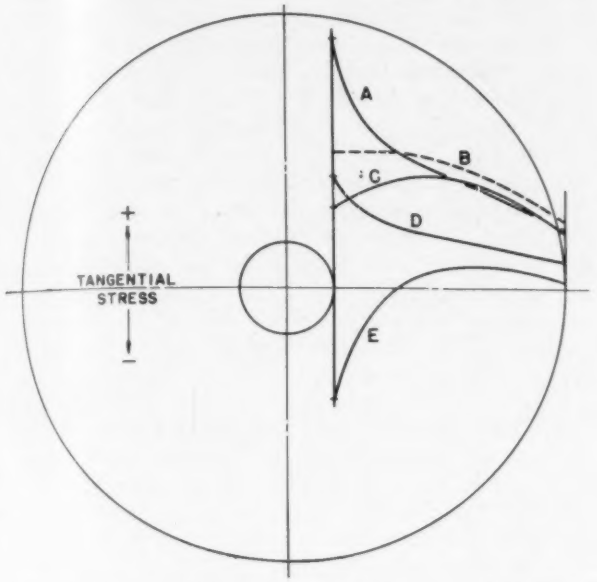
■ Fig. 4 – Concentric cracking in stresscoat, indicating compressive residual stress condition existed before drilling

■ Fig. 5 – Radial cracks, indicating tangential stress at central hole of rotating disc

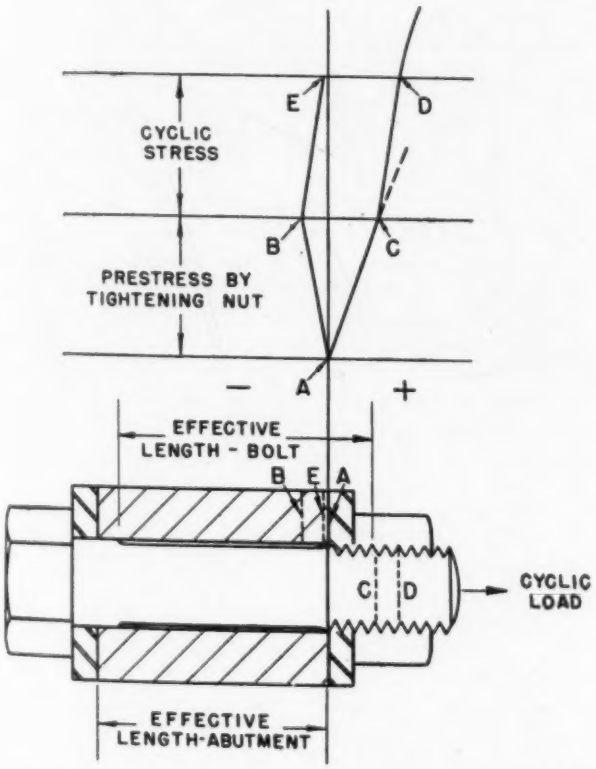
design, also in the fabrication of machine parts. Shot peening,[6] which sets up compressive stresses in the surface of a part, has proved its value in increasing the life of a part which is subjected to repeated tensile bending stresses. A beneficial effect can be obtained on machine parts by deforming the surface that is subjected to fatigue, by mechanical working. For example, the fatigue resistance of parts with transverse bores, keyways, screw threads, grooves, and fillets can be increased by plastically deforming the surface with suitable tools.

■ Prestressing During Assembly

An increase in the maximum allowable operating speed of rotating parts can often be obtained by so-called "overspeeding." For simplicity, a uniformly thick circular disc with a central hole will be used as an example. It is assumed that the thickness of the disc is relatively small in



■ Fig. 6 – Stress distribution curves, indicating method of setting up beneficial prestresses by overspeeding a rotating disc

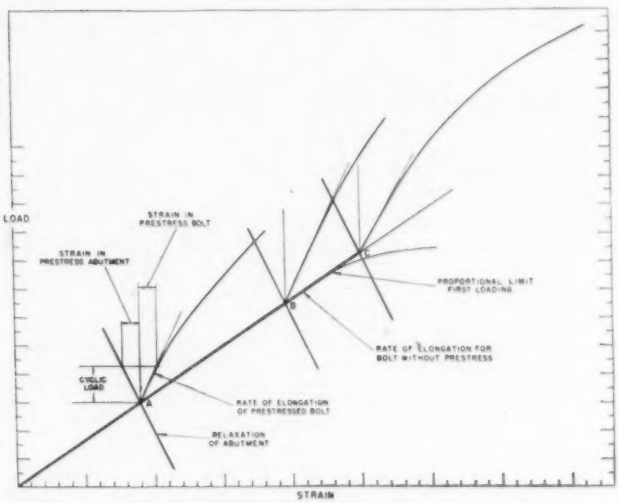


■ Fig. 7 – Illustration of phenomenon that takes place upon applying additional load to prestressed bolt

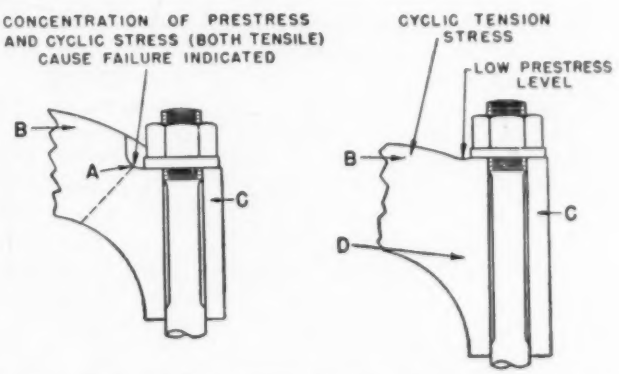
relation to its diameter and homogeneous material is used for its construction. Upon rotating this disc, tangential stresses are set up in the metal surrounding the central hole. Such a stress condition is shown in Fig. 5. This disc had been previously coated with stresscoat and then rotated. The cracks in the stresscoat indicating tangential stress show the highest strain to be located at the central hole in the disc. The tangential *elastic* stress distribution, symmetrical about the axis of the disc, can be illustrated by Curve *D* in Fig. 6. Upon increasing the rotative speed of the disc, a speed is reached at which the metal surrounding the central hole has been stressed above the proportional limit but below the ultimate strength of the material. The theoretical stress distribution for this condition, assuming elastic action, can be illustrated by Curve *A*. This stressing is not compatible with the yield strength and a more nearly correct distribution is shown by the broken line, Curve *B*. Upon stopping the disc, a tangential-residual stress condition will be set up. These residual stresses result from the permanent set of the material at the hole. The portion of metal which has yielded is compressed by the elastically strained outer portion and equilibrium is established by the tensile stresses set up in the outer portion.

The tangential residual stress, produced at the hole by stressing the material in the plastic range, is opposite in sign to the stressing produced by rotation of the disc. This condition produces a favorable effect on the final distribution of the stresses in the disc.

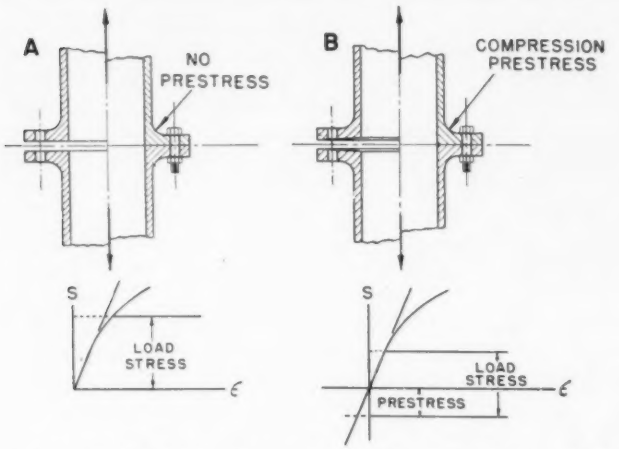
When the disc is again rotated, the tangential stressing



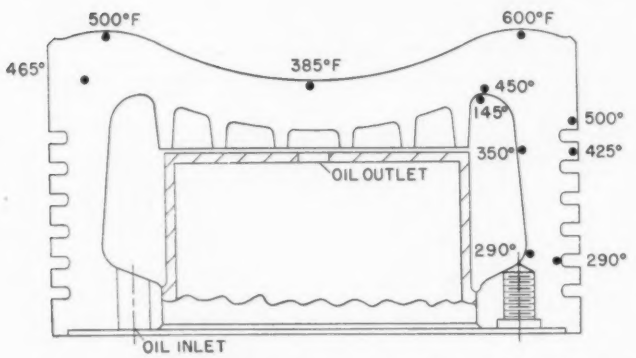
■ Fig. 8 – Effect of prestress on rate of strain with different amounts of prestress



■ Fig. 9 – Typical design feature which may cause premature failure due to concentration of prestress and cyclic stress at same location (left: former design; right: improved design)



■ Fig. 10 – Beneficial effects obtained by prestressing axially loaded member



■ Fig. 11 – Working temperatures found with 21½ in. diameter oil-cooled diesel-engine piston head (alloy cast iron)

in the disc may be obtained by the algebraic addition of the elastic Curve *A* and the residual Curve *E*. The stress distribution under this condition will be as shown by Curve *B*. The stress distribution for any speed below the proof speed may be obtained in this manner.

If the proof test speed is set to produce stressing as shown by Curve *B*, the stress distribution at the rated speed will be represented by the Curve *C*.

The maximum tangential stresses in Curves *C* and *D* are equal. If Curve *D* represents the tangential stress distribution at the maximum rated speed where the stresses are within the initial proportional limit of the material, the increase in speed as the result of overspeeding will be the difference in speed required to produce the stress distributions as shown by Curves *D* and *C*.

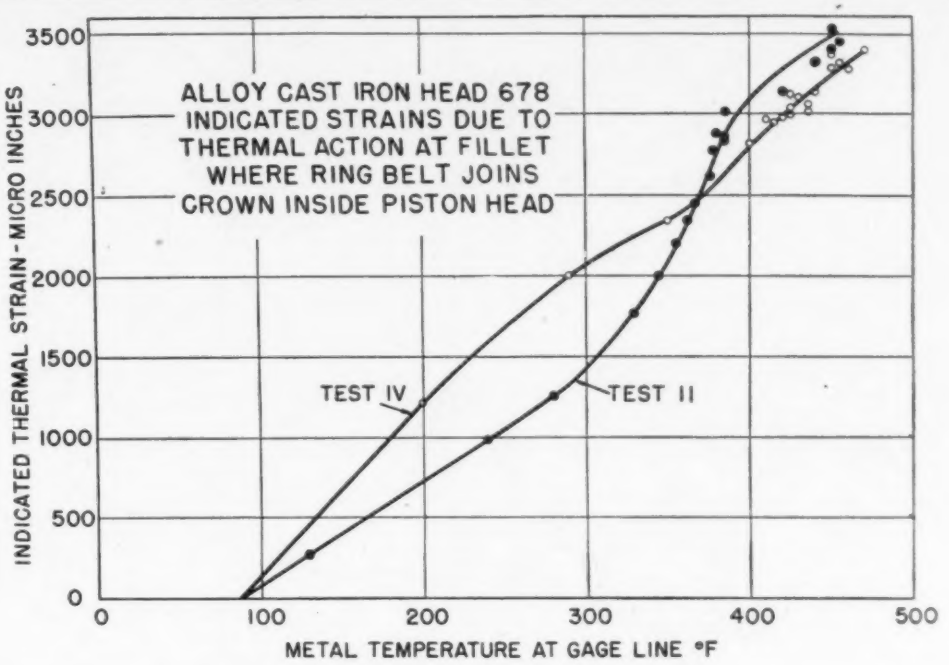
Upon increasing the rotative speed of the disc above the previous maximum speed, yielding will again take place and a new level of residual stresses will be set up on returning to zero speed.

No favorable effect will be obtained in overspeeding a rotative part unless the stress at some section of the part has reached a level above the proportional limit of the material.

Screw thread fastenings can be used to set up prestresses in assembly, whereby an improvement in the life of the parts can be obtained. Fig. 7, which shows a bolt and a tube, can be used to illustrate the phenomenon which takes place upon applying an additional load to a prestressed bolt. As the nut on the bolt is tightened, the bolt will elongate from *A* to *C* according to the modulus of elasticity of the bolt material, the area of the cross-section, length of bolt, and load. At the same time, the tube surrounding the bolt will shorten. This shortening will be governed by its modulus of elasticity, cross-section area, and effective length, or *A* to *B* as indicated in the stress-strain curve shown above the bolt illustration. When a load is now applied to the bolt as indicated by the arrow, it will elongate as *C* to *D* but, at the same time, the tube will also elongate as shown by *B* to *E*. The new rate of deformation with the prestressed bolt will be governed by the ratio of the effective cross-section of the tube, effective cross-section of the bolt, their effective lengths, and their respective moduli of elasticity. Changing any of the above design features will change the rate of strain in the bolt and tube when the assembly has been prestressed.

Generally, screw thread fastenings are prestressed in service. The amount of prestress should be governed by the service requirements. In Fig. 8 is shown the effect of prestress on the rate of strain for a bolt similar in design to Fig. 7. The data used in the construction of this diagram were obtained from a heat-treated steel bolt installed in an aluminum boss. Baldwin-Southwark Type SR-4 strain gages of ¼-in. gage length were used for the strain measurements. The line connecting the points *A*, *B*, and *C* indicates the rate of strain when the load is applied to the bolt without prestressing. The rate of strain in the bolt is approximately as indicated by the line *A* to *C*. The proportional limit in tension is indicated for the first applied load on the bolt. The nut on the bolt was tightened, setting up a strain in the bolt to the level as shown by the point *A*. Upon applying additional load to the bolt, the load-strain rate is less than the load-strain rate of the bolt without prestress. The rate of relaxation of the aluminum boss is shown intersecting the rate of strain in the bolt, at the prestress level. For a given cyclic load applied to the prestressed bolt, the cyclic strain in either the abutment or the bolt can be found as indicated in the figure. At points *B* and *C* are shown the effects when greater amounts of prestress are used. When using the amount of prestress as indicated at *C*, the cyclic load which can be added without plastic yielding is of relatively small amount. By prestressing the bolt above the proportional limit of the material and then loosening, residual stresses will be set up in the bolt at the points of stress concentration. This residual stress will be beneficial when the bolt is subjected to fatigue. The above case is an example where improvement in fatigue life can be obtained by prestressing in the same direction as the imposed stress in the bolt and, by so doing, decrease the stress range.

A feature of design which has been the source of a considerable number of fatigue failures where screw thread fastenings have been used is shown in the illustration at the left in Fig. 9. This type of design has been used particularly in highly stressed connecting-rod and main bearing caps. A spot face or counterbore has been used to provide a seat for the washer and nut. The fillet in this counterbore is generally located at the minimum cross-section of the cap as shown in the illustration at the point *A*. This is also the location of a high cyclic stress. Changing the design as shown at the right in Fig. 9 lowers the tensile prestress level, the combined stress, and improves stress conditions of the screw thread fastenings.



■ Fig. 12 – Indicated thermal strain at fillet between crown and ring belt inside piston. These strains found with piston head shown in Fig. 11

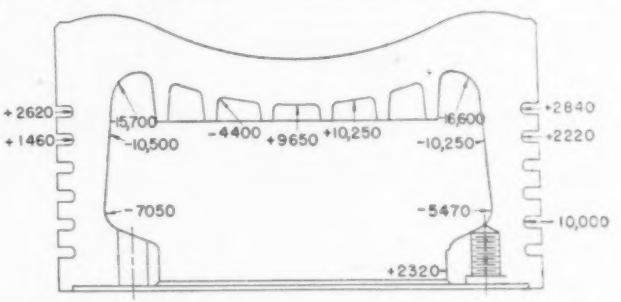
In Fig. 10 is shown a method whereby the prestressing may be set up in the opposite direction from the imposed stress to improve the service life. In both the illustrations *A* and *B*, the left half shows the machining details. The right half shows the parts held together with one of the bolts used in clamping the parts. Illustration *A* shows a conventional assembly in which the mating parts have been machined parallel. When clamped together, any prestress in the radius where the tube joins the flange is of a low level. In *B* the mating surfaces have been dished. Consequently, when these surfaces have been forced into intimate contact by the bolts in the flange, a prestress in compression exists in the radius where the tube joins the flange. Upon applying an axial load to this assembly, before a tension stress can be set up in this radius, the assembly must be loaded a sufficient amount to reduce the compressive prestress to zero. This is shown in the stress-strain diagrams. If strain gages have been applied at the radius in both designs, it would be found that the strain readings would be approximately equal for a given load. However, the strain readings obtained from assembly *A* would be strain values representing tensile stress, while the strain readings obtained from assembly *B* would represent a strain range resulting from some value of compressive prestress. To evaluate the tensile stress, it would be necessary to determine the compression strain set up at gage location due to the tightening of the flange bolts, and then subtract this value from the value obtained under load in order to determine tensile strain.

■ Thermal Stress

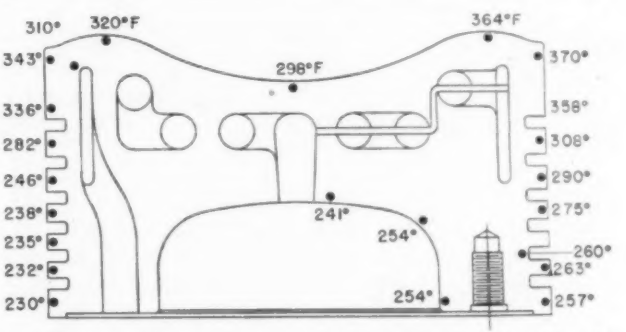
Thermal stresses are induced by restraining the normal expansion or contraction which would accompany a change in temperature. A very complex system of stresses can be set up within a part by the temperature gradients which generally prevail at elevated temperatures.

Fig. 11 shows the temperature distribution which prevailed under simulated operating conditions in a 21½-in. diameter oil-cooled alloy cast iron piston head used in a diesel engine. The lower end of the ring belt offers considerable restraint to the crown as well as to the upper part of the ring belt, and a high tensile thermal strain is set up at the fillet where the ring belt joins the crown.

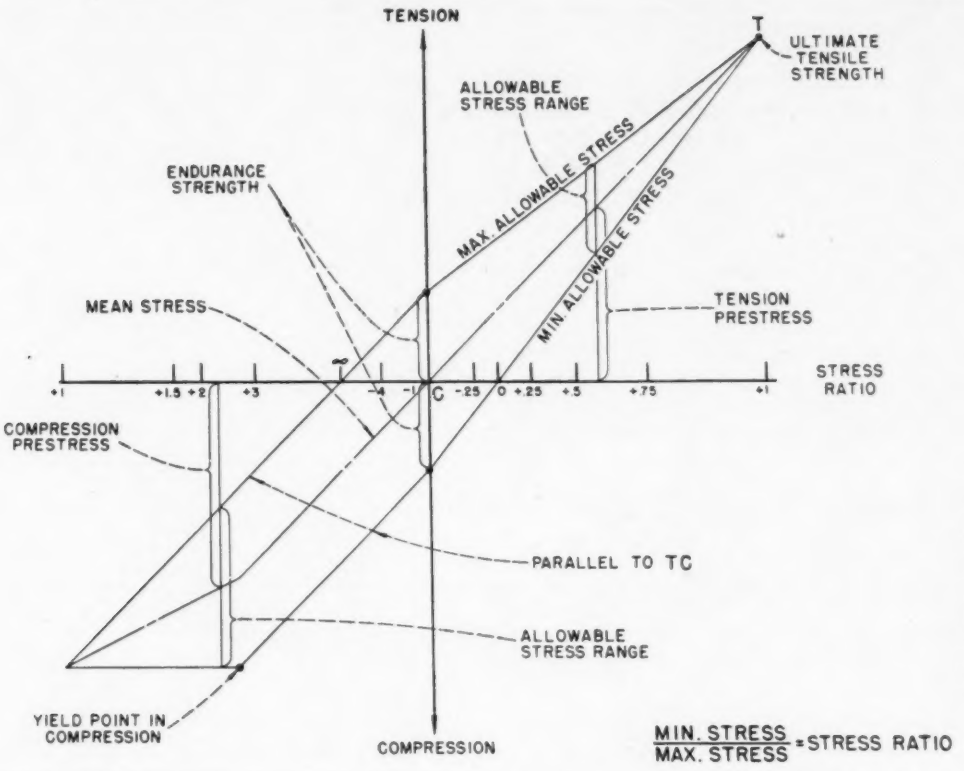
Fig. 12 shows the variation with increasing temperature in thermal strain at the fillet in this piston head. These strains are reproducible as shown by the results from two



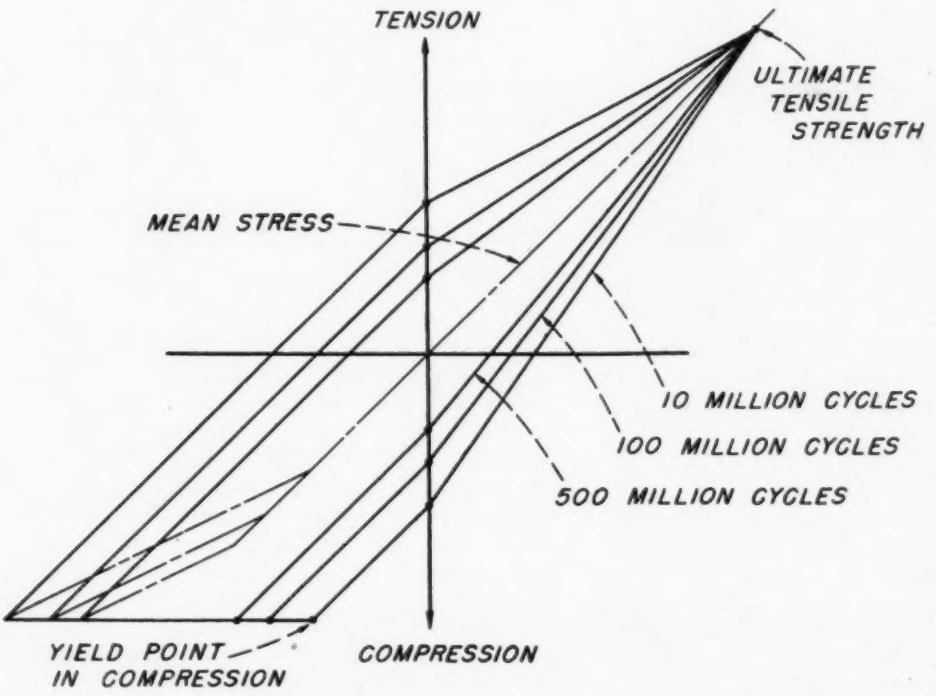
■ Fig. 13 – Indicated cyclic stress range due to explosion pressure. Operating temperatures of this piston shown in Fig. 11



■ Fig. 14 – Working temperatures of aluminum alloy piston head. Design proposed to replace piston head shown in Fig. 11



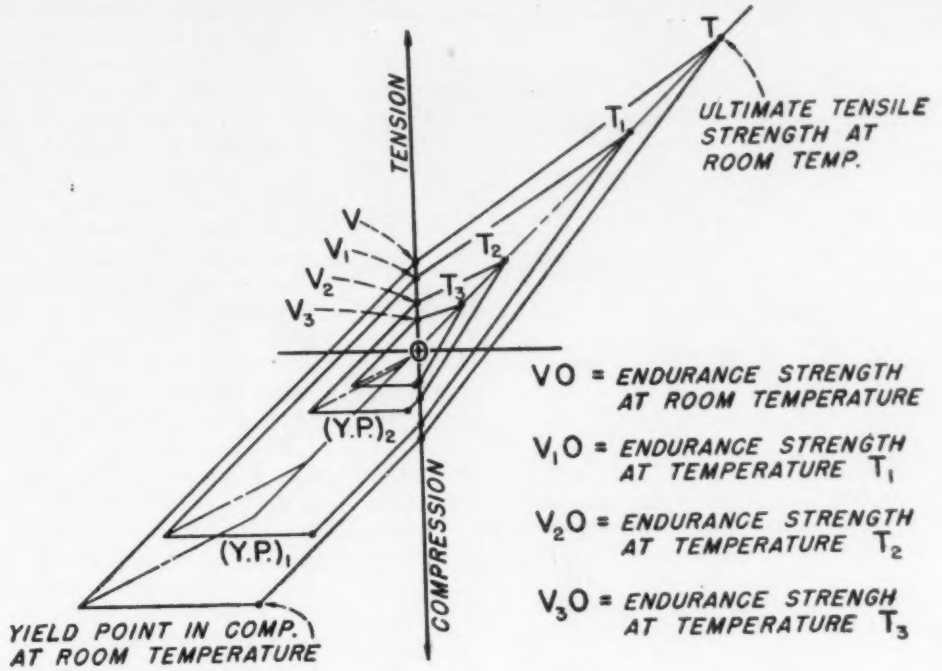
■ Fig. 15 – Modified Goodman diagram of fatigue strength of aluminum alloy at given temperature and for given number of cycles (notations indicate method of construction)



■ Fig. 16 – Diagram of fatigue strength of aluminum alloy at given temperature for various numbers of cycles

of several tests. From tensile test it was found that the ultimate strength and modulus of elasticity of this material up to 450 F were approximately 48,000 psi and 20,000,000 psi, respectively. When this modulus is multiplied by the strain of 3500 microin. (shown in Fig. 12 at 450 F), the apparent stress is 70,000 psi. This strain must be in the plastic range, as the apparent stress is not consistent with the ultimate strength of the material.

Fig. 13 shows the range of the imposed stress at various gage lines on the piston head resulting from a simulated explosion pressure of 1500 psi in the combustion chamber. Two gages mounted at right angles were used at each gage line. The imposed stress has been computed from the strains indicated at each gage line, by the formulas shown in Fig. 3. These imposed stresses are repeated with each explosion and are algebraically added to the



■ Fig. 17 – Diagram of fatigue strength of aluminum alloy at elevated temperatures for given number of cycles

stress resulting from the thermal gradient.

Provided no stress other than the thermal stress is present in the piston head at the fillet, the imposed stress of −16,600 psi would be applied to a high level of tensile prestress resulting from the thermal gradient. The stresses would combine as the algebraic addition of imposed and thermal stresses. This means that in this instance the imposed stress range would be between two levels of tension. A fracture could be expected at a low number of cycles due to the relatively high level of pulsating stress.

A proposed design of an aluminum piston head is shown in Fig. 14. The temperatures indicated in this figure were obtained with the piston head subject to the same operating conditions as prevailed with the test of the piston shown in Fig. 11. This aluminum piston head gave considerable improvement in operating temperature, temperature gradient, imposed stress range, thermal stresses, together with about 40% saving in weight.

■ Combined Stress

Residual, thermal, machining, and assembly stresses generally function as a prestress upon which is applied the stress resulting from the operating load. Stresses cannot be computed vectorially, as is possible when dealing with forces. Careful analysis should determine the prevailing stress level.

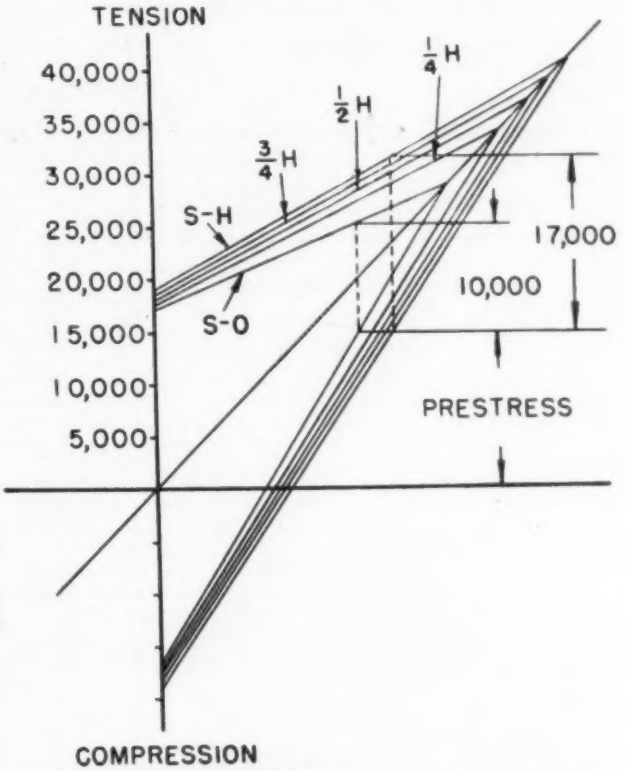
■ Allowable Stresses

Having determined the stresses and stress ranges in the machine part, consideration must be given to the properties of the material to determine the strength or service life.

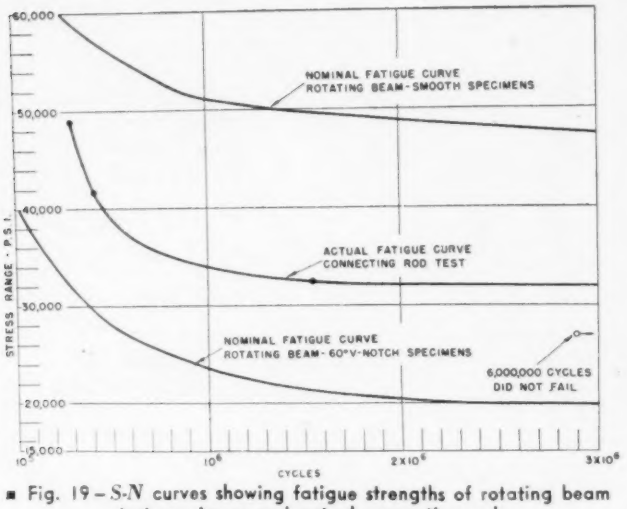
For static loading, it is relatively simple to determine the safe allowable working stress. Most engine parts are subject to repeated stressing; therefore, the resistance to fatigue greatly influences the service life.

Fatigue strength values as determined by the usual laboratory tests cannot be used directly because such tests are usually performed on specimens carefully prepared to reduce to a minimum the effect of mechanical stress-raisers, such as grooves or scratches.

The comparative endurance or fatigue strengths of vari-



■ Fig. 18 – Diagram showing allowable range in stress for specific minimum stress increased by cold working (52S wrought aluminum)



■ Fig. 19 – *S-N* curves showing fatigue strengths of rotating beam test specimens and actual connecting rod

ous materials as usually quoted refer to rotating beam or bending tests on relatively small specimens in which complete reversal of stress takes place each cycle. The value quoted for a given number of cycles is the computed stress in the specimen at the time of failure. The stress range is twice the fatigue strength value as given for rotating beam or bending tests. The stress ratio is the algebraic minimum stress divided by the algebraic maximum stress. Thus, complete reversal of stress is referred to as a −1 stress ratio. Rarely in service are machine parts subject to complete reversal of stress. Parts in service may have a stress ratio anywhere between plus ∞ and minus ∞.

The nominal allowable cyclic working stress range in tension, compression, or various combinations thereof can be obtained from a modified form of the Goodman stress range diagram. The method of constructing this diagram is indicated by the notations in Fig. 15. The data needed for its construction are the ultimate strength, the endurance limit or fatigue strength for reversed bending or direct stress, and the yield strength in compression – all for a given temperature.

The shape of this diagram is for aluminum alloys – other materials may or may not have a similar diagram. Fig. 16 shows the effect on the diagram of various numbers of cycles, while Fig. 17 shows the effect of elevated temperatures.

If the material will be cold worked before use in the part, a new diagram should be constructed as the properties of the material have changed.

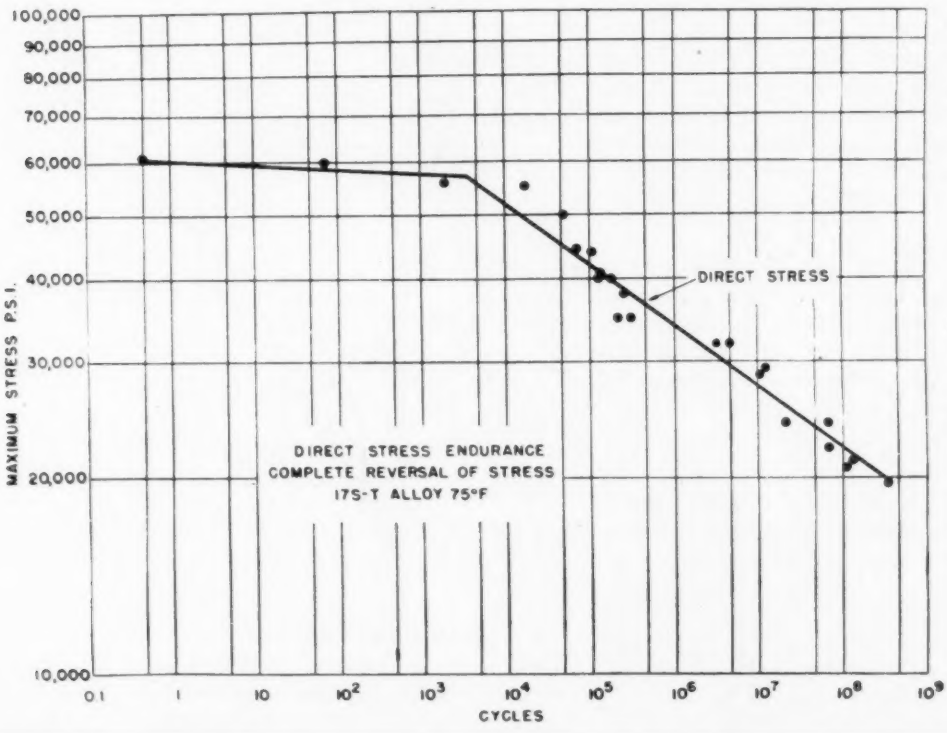
An example of the effect of cold working an aluminum alloy is shown in Fig. 18. In this figure, the endurance strength and ultimate strength have been plotted as a diagram showing the allowable working strength under repeated loading. It can be seen that considerable increase in allowable range in stress for a specific minimum stress can be obtained by cold working, although the fatigue strength under complete reversal shows a comparatively small per cent of increase.

When the allowable repeated stress range diagram has been constructed using the typical mechanical properties of material and fatigue data from rotating beam test specimens, the stress range obtained must be corrected for stress concentration as found in published literature, such as Roark's work.[7] If the diagram has been constructed using the properties obtained from tests of actual machine parts under direct stress, lower safety factors may be used.

■ Fatigue Strength of Machine Parts

The possibilities of designing a machine member free

[7] See "Formulas for Stress and Strain," by R. J. Roark. Published by McGraw-Hill, New York, 1943.



■ Fig. 20 – Log-log plot of *S-N* curve for wrought-aluminum alloy for range between 0.5c and 500,000,000c

of stress concentration are extremely limited. Machine parts will generally show a resistance to fatigue somewhere between the stress causing failure in a smooth finish test bar and the notch fatigue test bar. The design may approach perfection but it will still be necessary to allow for the variation of properties resulting from manufacturing tolerances.

Fig. 19 shows an *S-N* curve in which the stress range, instead of maximum stress, as is the usual procedure, has been plotted against cycles to failure. The stress ratio used for the connecting rod test was minus 2. The curve at the top is the allowable range of stress for a minus 2 stress ratio as indicated from rotating beam tests where the test specimen has been smoothly polished. The curve at the bottom is also from a rotating beam test (stress range indicated for minus 2 stress ratio), but in this case the test specimen had a 60-deg sharp notch machined in the test section. The curve between these two curves has been plotted from results obtained from actual connecting rods fabricated from the same alloy as the rotating beam test specimens, in which the stress range under test conditions has been obtained from strain measurements.

It would not be reasonable to expect that the material in a large machine part would be as free of stress-raisers as a laboratory test bar; thus, due allowance must be made in the allowable stress range. Some engineers have referred to the differences in fatigue results between actual machine parts and laboratory test bars as "size effects."[8] The accepted definition of size effects is the variations in fatigue strength with specimens of different sizes but of the same general shape and metallurgical structure tested under repeated stress. The differences in fatigue properties between laboratory test bars and machine parts are caused by variations in metallurgical structure and stress-raisers. This difference may result from many causes and space does not permit a full discussion in this article.

The question pertaining to when fatiguing action starts has often arisen in discussions. The answer is not important but it is important that the engineer have available complete fatigue data on the material in question. Such a curve is shown in Fig. 20. This curve shows a log-log plot of an *S-N* curve for a wrought aluminum alloy for the range between 0.5 c and 500,000,000 c. With this material up to approximately 10,000 c, the allowable repeated stress is above the yield strength of the material as determined by conventional test. After this number of cycles, the well-known *S-N* curve exists. The fatigue strength at elevated temperatures is complicated by aging effects with consequent changes in properties.

Fig. 21 shows *S-N* curves plotted log-log for a typical cast-aluminum alloy. The rotating beam curve has been plotted as *stress range* for a stress ratio of minus 1. The other curves have been plotted from results under direct stress tests using specimens having a ½ x 8-in. test cross-section with the surface of the test section in the as-cast condition. The curve labeled "fully reversed" was obtained from specimens tested with a stress ratio of minus 1. The curve identified as "all compression" is from results obtained under a stress ratio of infinity, while the curve "all tension" results from specimens tested with a stress ratio of zero. The range of stress in compression (infinity stress ratio) must be limited to the yield strength in compression. Failure by fracture may not take place at some stress levels greater than the yield strength in compression; nevertheless, many parts could not function properly due to the permanent distortion which would take place.

The fatigue strength as indicated for the ½ x 8-in. test specimens, tested under direct stress, may compare with the fatigue strengths to be expected in the web sections of a cast crankcase.

Mechanical properties obtained from test specimens removed from machine parts are generally below the mechanical properties obtained from laboratory test bars of the same alloy and heat-treatment. Examples of factors causing this difference are grain flow in forgings and rate of solidification in castings. In addition, we have the influence of surface irregularities, internal discontinuities, and abrupt changes in section, all of which greatly influence the strength of a material.

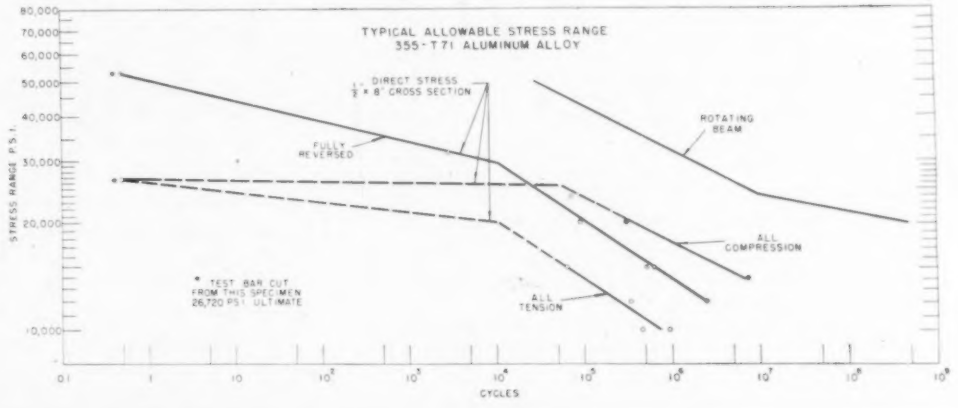
■ Summary

A few of the many methods whereby the life of machine parts can be improved have been presented. Many failures of machine parts in service have been attributed to material. A great percentage of these failures should be charged to design. Machine parts generally contain a sufficient amount of material, although often the material is inefficiently distributed.

The criterion of the strength of a design should not be

concluded on page 502

[8] See Proceedings of Society for Experimental Stress Analysis, Vol. 2, No. 1, 1944, pp. 170-177; "Study of Residual Stresses and Size Effect and Study of Effect of Repeated Stresses on Residual Stresses Due to Shot Peening of Two Steels," by H. F. Moore.



■ Fig. 21 – Fatigue curves of cast-aluminum alloy plotted log-log for stress ratios of plus 1, minus 1, and infinity for the range between 0.5c and 500,000,000c

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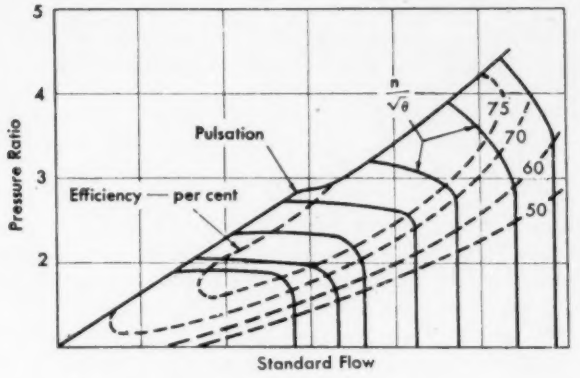

■ Fig. 1 – Centrifugal compressor performance map

THE centrifugal compressor and the gas turbine have been closely related throughout the period of their history, and the development of each has been largely dependent on that of the other. One of the early gas turbine experiments by Armengaud and Lemale for the Société Anonyme des Turbomoteurs in Paris employed a multistage centrifugal compressor built by Brown-Boveri. This compressor was the first multistage unit attempted by that company, and 25 stages were used to obtain an overall pressure ratio of about 4:1, which corresponds to a pressure ratio per stage of approximately 1.06:1. The General Electric Co. built its first centrifugal compressor in the Lynn Works in 1903. This was an experimental, single-stage compressor which developed a pressure ratio of approximately 1.3:1, and it also was a component in gas turbine experiments being conducted by the General Electric Co.

Perhaps the earliest practical application of the gas turbine is found in the turbosupercharger used to increase the output of the otto cycle and diesel engine, and to effect altitude compensation for the conventional aircraft engines. Although improvements in the centrifugal compressor had been made since the first experimental models the need for larger pressure ratios and smaller, lighter compressors for effective turbosupercharging gave impetus to the development of the centrifugal compressor, particularly with respect to these characteristics, and the pressure ratio obtainable from a single stage rapidly increased. The General Electric Co. in Lynn has recently obtained pressure ratios exceeding 6:1 from experimental, high-speed centrifugal compressors. A general evaluation of the present state of development of the centrifugal compressor is indicated briefly in the following tabulation of presently obtainable values of pressure ratio, efficiency, and flow for single-stage machines:

Compressor Pressure Ratio	Compressor Flow Q/nD^3	Compressor Efficiency
6:1	0.14	0.70 (minus)
4:1	0.14 to 0.17	0.75 to 0.70
2:1	0.16 to 0.35	0.75 to 0.80 (plus)

The high-speed, single-stage, centrifugal compressor is well suited to the needs of aircraft gas turbines in many respects. It is a simple, compact, light in weight, easily manufactured compressor having adequate performance for many applications.

THE AUTHOR: R. S. HALL, group leader for compressor research work at General Electric Co., has been associated with GE since 1936. His work there has included drafting, testing, routine engineering calculations, and research work in the newly established Supercharger Division. He finally concentrated his efforts on the Supercharger Engineering Division (now the Aircraft Gas Turbine Engineering Division) and has held several different functional positions in the division.

■ Balancing Design Factors

In achieving full realization of the inherent advantages of the centrifugal compressor in an aircraft gas turbine it is necessary to appraise the relative magnitude and importance of many factors. These factors fall quite naturally into a few general categories: application requirements, mechanical design considerations, and thermodynamic or fluid-flow considerations. The balancing of the important factors under the mechanical and application classifications is discussed in other papers on these factors, as well as certain of the thermodynamic considerations. In addition to the aspects of aircraft gas turbine design being treated in these papers there is required a complete understanding of the effect of the various components on the off-design operating condition.

■ Off-Design Condition

An adequate treatment of this problem requires a knowledge of the performance characteristics of the compressor, turbine, and combustion chamber over the useful range of speed and flow. Assuming that such data are available it is possible to evaluate the magnitude and relative importance of the factors controlling the off-design condition by means of a simply constructed performance map. The method now used to represent compressor performance is an example of a performance map. Fig. 1 is such a representation of performance for a typical high-speed, single-stage, centrifugal compressor. It shows parameters of speed and efficiency plotted against pressure ratio and flow as ordinate and abscissa in terms of standard conditions, which is basically a form of dimensionless representation.

This compressor performance map is a suitable basis for a complete gas turbine map, and the problem of analyzing the off-design operating condition can be regarded as that of mapping the desired gas turbine parameters in the compressor performance field.

The desired mapping of the turbine parameters into the

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TURBINES with CENTRIFUGAL COMPRESSORS

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compressor performance field is easily accomplished with the aid of three fundamental relationships which must be satisfied:

1. Compressor rpm = turbine rpm.
2. Compressor flow plus fuel flow = turbine flow.
3. (a) Jet engine: compressor power = turbine power. (b) Propeller-drive engine: compressor power + propeller power = turbine power.

■ Design Point Location

The first step in the solution is the establishing of the design point. This is determined by considering the results of cycle analyses in the light of practical limitations and results in establishing the pressure ratio, flow, and gas temperature at this point.

The location of the design point in the compressor performance field is thus given and consequently the compressor performance. As the compressor map, as shown in Fig. 1, has no representation of power, it is necessary to add a parameter for indicating this quantity. The power is most conveniently represented by specific power; that is, power per unit airflow. This parameter has been added to the compressor map in Fig. 2 and is denoted by $hp/W\theta$. Fig. 2 also shows (small circle) the location of the design point.

Condition No. 1 – The fulfillment of the requirement that the compressor and turbine speed are equal is automatically satisfied by virtue of direct mechanical coupling; consequently the speed parameter on the compressor map serves equally well for the turbine.

Condition No. 2 – The fulfillment of the requirement that the compressor flow and fuel flow equals the turbine flow is satisfied by the following process. The fixing of the design point determines the areas and vector relationships for the turbine, so that the problem of determining the flow at conditions other than those for the design operating condition is a simple solution for the flow through fixed areas for various pressure ratios and gas temperatures. In order to "map" the turbine flow into the compressor field, it is necessary to refer the flow parameter to the same reference temperature used for the compressor. This is accomplished by plotting pressure ratio against flow for constant values of turbine inlet total temperature divided by compressor inlet total temperature, in terms of compressor pressure ratio. The value of pressure ratio plotted then becomes the turbine pressure ratio increased by the pressure ratio across the combustion chamber.

Condition No. 3 – The turbine shaft efficiency is determined by the pressure ratio and velocity ratio. As the velocity ratio is given by the wheel speed divided by the velocity equivalent of the available energy, the pressure ratio, speed, and gas temperature are sufficient to determine the turbine shaft efficiency. The turbine shaft efficiency and the available energy determine the turbine power; consequently, lines of constant turbine specific power can be mapped. Fig. 3 shows the turbine parameters of flow and power determined in this fashion.

The fulfillment of the third requirement consists of locating the points where the compressor and turbine power are equal and is found at the intersection of the turbine and compressor power lines of the same value. This process is indicated in Fig. 4.

The resulting complete gas turbine performance map is shown in Fig. 5.

■ Range of Applicability

It is important to consider the accuracy of this representation of performance in order to evaluate its range of applicability.

■ Similitude Considerations

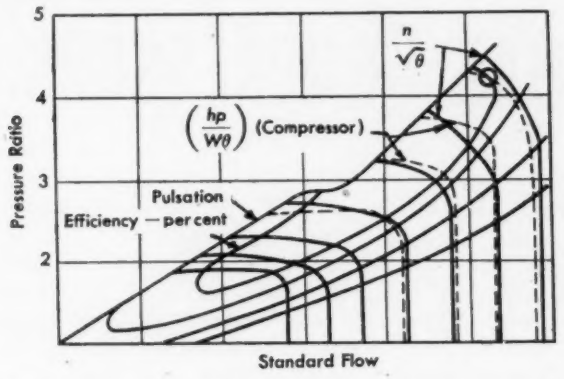
The representation is fundamentally a dimensionless one and can be evolved through considerations of dynamic similitude. There are, however, several qualifying considerations. First the effects of Reynolds number and heat transfer are assumed to have negligible effect. The effects of these approximations are of small magnitude throughout the operating range and no appreciable effect should be expected at least within limits of ordinary engineering accuracy.

■ Combustion Chamber Effects

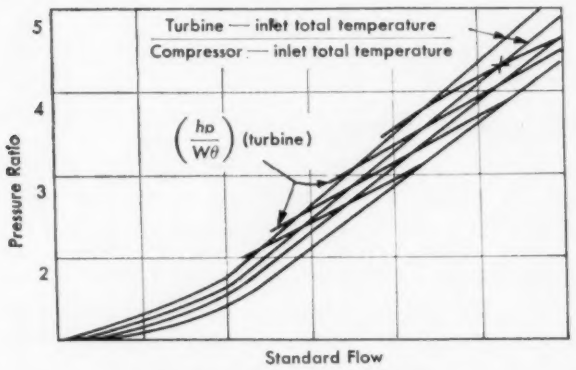
A second factor which can affect the accuracy is found in the combustion chamber. The combustion efficiency is affected by the absolute pressure. A more complete discussion of this phenomenon is not included here, but the effect upon the validity of the performance map is important here. The effect can be separated into two components. The first does not affect the accuracy of the performance map but applies only to calculations of fuel consumption and amounts simply to increasing the fuel

ALONG with further improvements in the form of increased flow and pressure ratio for gas turbines with centrifugal compressors, the author looks forward to an improvement in efficiency, this latter factor being more important in gas turbines than in aircraft turbosuperchargers.

As an aid to the realization of this aim, the author presents here a method of determining approximate gas turbine performance at various operating conditions.



■ Fig. 2 – Compressor performance map with power required per unit airflow



■ Fig. 3 – Turbine flow and power

consumption by a factor, described as combustion efficiency, over the value determined from the map. The second component is negligible except at operating conditions where very low combustion efficiency obtains. In this region the turbine flow must be corrected by the corresponding increase in fuel-air ratio.

■ Effect of Altitude

The third consideration in evaluating the limitations of the performance map involves the effect of the external air conditions, that is, the pressure and temperature of the air surrounding the engine. Other than the effects already considered, the map is unaffected by changes of the pressure and temperature level. This is ensured by satisfying the requirements of dynamic similitude which gives the dimensionless form of representation or, its equivalent, the expression of the various parameters in terms of standard conditions.

■ Effects of Airplane Mach Number

The remaining consideration consists in evaluating the effects of ram or airplane speed (Mach number). There are three significant phases of the effects of ram:

1. When the jet nozzle effective pressure ratio is above critical.

2. When the jet nozzle effective pressure ratio is below critical, but the turbine nozzle pressure ratio is above critical.

3. When both the turbine and jet nozzles are below critical.

Within the region where phase (1) exists there is no effect on the relation between the turbine pressure ratio and flow and that for the compressor, and the map is unaffected.

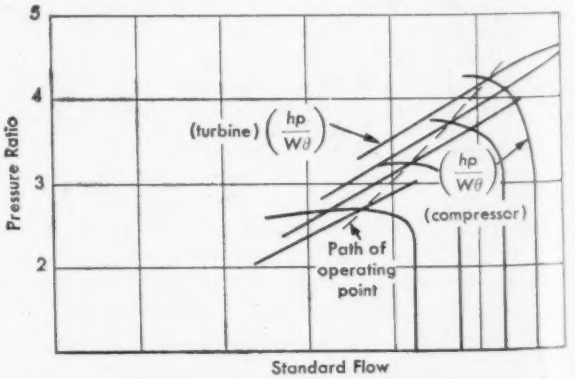
Within the region where phase (2) exists the turbine pressure ratio is altered while the compressor pressure ratio remains unaffected. The shift in turbine pressure ratio affects its power and the operating point is shifted.

Within the region where phase (3) exists both the turbine flow and pressure ratio are altered in relation to the compressor, with the result that the operating point shifts.

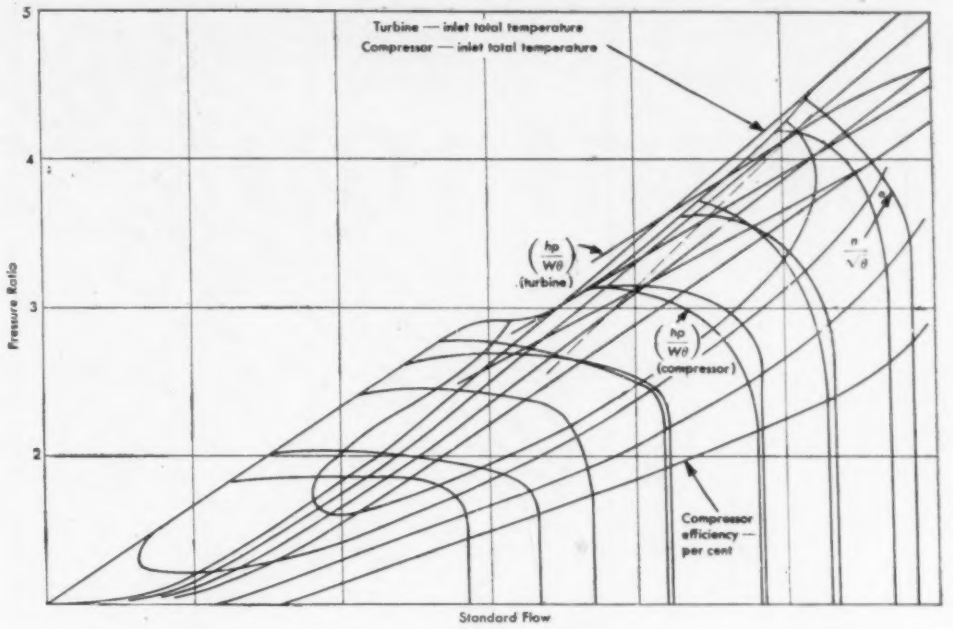
The conditions under phases (2) and (3) can be handled by a refinement in the performance map consisting of plotting the turbine parameters of Fig. 3 on a transparent overlay with logarithmic scales used for both the compressor and turbine ordinate and abscissa. Phase (2) is then handled by a shift of the turbine map along the direction of the ordinate by an amount corresponding to the change in effective pressure ratio. Phase (3) can be handled in a similar fashion, but requires an additional shift along the abscissa by an amount corresponding to the change in the flow relationship. This refinement is approximate to the extent that the effect of the change in turbine velocity ratio on the turbine efficiency is not considered, which results in small effect in the useful operating range, as the turbine efficiency curve is reasonably flat in this region.

■ Jet Engine

The steady-state operating condition for the jet engine is shown in Fig. 5 as the dash line. It is important to notice the change in location of the operating point induced by the acceleration and deceleration phase of operation. This is easily done by locating the intersection of a given compressor power line with an appropriately higher or lower valued turbine power line to correspond with the power absorbed by the inertia of the rotor. The general effect is that the operating point moves along a constant compressor power line toward reduced flow and increased gas temperature for acceleration and toward increased flow and reduced gas temperature under deceleration. These effects broaden the path of the operating point to a band of some width, which, at high rotative speeds, particularly at high altitude (where $n/\sqrt{\theta}$ is larger for a given rotor rpm), may intersect the pulsation region during accelera-



■ Fig. 4 – Path of operating point



■ Fig. 5 – Complete gas turbine performance map

tion unless care is taken in determining the location of the design point in the compressor performance field.

■ Propeller-Drive Engine

The performance map will serve effectively to follow the additional effects introduced in the case of the aircraft gas turbine with propeller drive.

As the location of the operating point is dependent on the power being absorbed by the propeller, which is determined by plane speed and propeller pitch, it is desirable to direct attention to the effect on the gas turbine performance introduced by various propeller-pitch control schedules. The steady-state conditions are of paramount interest in this discussion; accordingly, the transient effects, which are primarily control problems, will not be considered here.

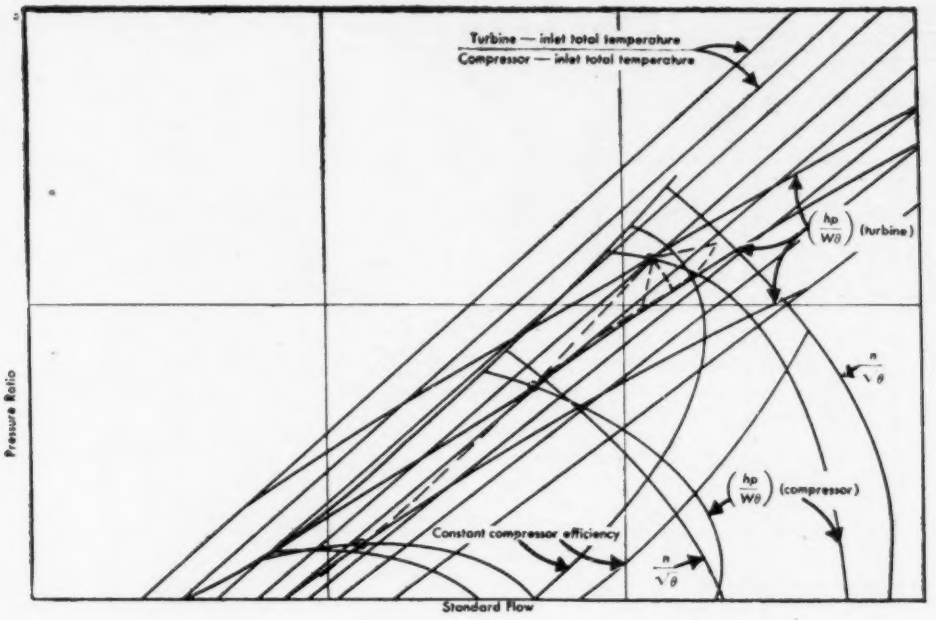
Fig. 6 shows the performance map for a propeller-drive engine on a somewhat larger scale and including only the region of interest in connection with normal operation.

Considering first the possibility of operating with a fixed-pitch propeller, it is assumed that the design point is determined for a particular altitude and airplane speed. If we assume that it is possible for the airplane to fly at such a speed and altitude that the propeller operates at a constant value of J or V/nD at constant altitude, the operating point will follow a path corresponding closely to that for the jet unit, as indicated by the long dashes. As the airplane actually will require more power to fly at the altitude and speeds required to fulfill this condition, it will be necessary to consider the effect of changing propeller J. If it is possible to operate a propeller-drive unit with a fixed-pitch propeller, it must be possible to reduce the J value to zero corresponding to zero plane speed. As the ratio of the power coefficient at J equals zero to that at J equals one is approximately 1.5:1, it is apparent that if the design point is taken at conditions of full plane speed, it will be impossible for the gas turbine to operate in the stable compressor region with more than approximately 60% of design output; on the other hand, if the design point corresponds to static conditions (J equals zero) the output at full plane speed will not be much over 60%. It is apparent that the airplane gas turbine with propeller must incorporate some additional flexibility in the system, such as a variable speed ratio between propeller and turbine or a variable-pitch propeller.

With a variable-pitch propeller the system can be adjusted with complete freedom as regards the location of the operating point on the performance map. It is, therefore, of some interest to notice the effects of controlling propeller pitch in accordance with several different schedules. In order to determine a range of possible schedules it is well to determine the location of a line of constant total power available to the propeller on the performance map. Such a line is indicated in Fig. 6; the dotted line approximately parallel to the turbine specific power lines locates the position of the operating point for operating conditions which will deliver to the propeller 80% of the output at the design point. Several paths connecting the design point with the 80% output line are also indicated in Fig. 6. These several paths correspond to controlling propeller pitch to maintain constant values of:

1. J for propeller
2. Airflow through gas turbine
3. Gas turbine speed
4. Gas turbine pressure ratio.

We notice that in all cases the gas temperature is reduced at reduced output, but that the speed decreases for (1) and (2), remains constant for (3), and increases for condition (4). A calculation of the specific fuel consumption for conditions along the 80% power line will indicate an optimum value for this particular turbine and compressor which lies in the region between the intercepts of the constant-speed line and the constant propeller J-factor line, and which is roughly along the constant-flow lines. The increase in specific fuel consumption at the other conditions is, however, quite nominal, being a maximum increase of about 2½% at the intercept of the constant pressure-ratio



■ Fig. 6 – Performance map for propeller-drive gas turbine

line. From these general considerations it is possible to draw some interesting conclusions; first, that a simple, fixed-pitch propeller directly coupled to a simple gas turbine will not operate satisfactorily; and second, that the deciding factor in determining the optimum schedule of propeller-pitch change is probably more a function of turbine life and control complexity than it is a function of efficiency. A more general effect is also noticed; that is, that the compressor pulsation imposes little or no limitation on the possible range of operation under part-load or off-design conditions.

■ Probable Trends

The future of the gas turbine with the centrifugal compressor is somewhat dependent on the probable direction of development of the centrifugal compressor. Past history shows that the development of the centrifugal compressor has been closely linked with the gas turbine. The advances in design methods and performance have been directly related to the practical requirements. For example, the demand for lightness and simplicity in the aircraft turbo-supercharger led to rapid increase in the pressure ratio obtainable in the single-stage compressor, but as the premium on efficiency was a relatively minor factor, little improvement in this characteristic was produced. In the rapidly developing field of the gas turbine there is a much greater premium on efficiency and it seems probable that, along with further improvements in increased flow and pressure ratio, we can expect important improvements in the efficiency, particularly in view of the fact that there is no known fundamental law which demonstrates that the centrifugal compressor should be basically less efficient than other types of turbo machines, such as turbines, fans, and axial-flow compressors.

DISCUSSION

Presents Variation Of Author's Method

—J. E. TALBERT

Wright Aeronautical Corp.

THE Wright Aeronautical Corp. has for some time used a similar method to estimate performance of a centrifugal compressor and a reciprocating engine. However, we have used a combination of Fig. 1 and a figure similar to Fig. 3 but based on airflow and density into the cylinder of the reciprocating engine. We do not use the horsepower parameter since we are more interested in obtaining the intake pipe density at any operating point, and use it with engine performance curves to obtain engine horsepower. We feel that use of the horsepower parameter would be too inaccurate.

In the paper, Figs. 1, 2, and 3 are carried further into Figs. 5 and 6. Although these latter figures are useful in obtaining approximate matching characteristics of the compressor and turbine, they are inaccurate in estimating net horsepower for a propeller-turbine engine. The interpolation between the lines is difficult because the net horsepower is the small difference between the two large powers of the turbine and compressor. The use of Fig. 5 for jet engines only gives the matching points and does not give the net energy in the jet. This must be calculated from the airflow, pressure ratio, and horsepower at each of the operating points. Since the net energy must be calculated anyhow, it would seem more accurate to use only curves 1 and 3 to obtain matched airflow and pressure ratio and use these numbers to calculate a net energy or horsepower.

We should note that the curves of Fig. 3 are really based on a pressure entering the turbine at least above critical pressure ratio. As the paper points out, they are only converted to a compressor pressure ratio ordinate by mathematics at a standard atmospheric pressure. The curves not only vary with airplane ram, as stated by the author, but also with a variation in area of a variable area jet nozzle. Corrections for these factors could more readily be applied to a simplified (no horsepower parameter) Fig. 3 than to Figs. 5 or 6.

Thus, we would like to suggest that Figs. 1 and 3 are more useful than Figs. 5 and 6 in accurately calculating the net energy output of a gas turbine.

TESTING OF HIGHLY LOADED SLEEVE BEARINGS

by J. PALSULICH
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and
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Wright Aeronautical Corp.

THIS paper describes a test program being conducted for the evaluation of highly loaded sleeve bearings. It lists the specific properties of bearings being investigated and describes in detail the test equipment and procedure used. Curves and tables are included showing the test results obtained.

The usefulness of laboratory test rigs in accelerating the development of new bearing materials and designs is stressed by the authors, who point out that the labor and expense involved in testing bearings is greatly reduced by the use of such rigs.

THE exacting performance requirements of aircraft engines, under conditions of increased power output and higher diving speeds with substantially fixed basic weight and size, have necessitated the development of sleeve bearings with high load-carrying capacity, reliability, and extended service life. In the field of development of highly loaded bearings, there are no established criteria to act as guides and the experience in other industries is not often of much value.

In order to reduce the time and expense involved in the testing of bearings in full-scale engines, the Wright Aeronautical Corp. has established an extensive supplementary program of bearing testing in laboratory rigs. Although it is recognized that the overall performance of a bearing can only be evaluated in full-scale engines operating under service conditions, these laboratory test rigs have been extremely useful in the evaluation of specific properties of new bearing materials, designs, and lubrication systems. The use of rigs has not only accelerated bearing development but greatly reduced its cost. This paper describes the testing of highly loaded sleeve bearings in laboratory test rigs and full-scale engines.

■ Properties of Bearings Being Investigated

1. Maximum load-carrying capacity.
2. Friction and lubrication characteristics.
3. Fatigue life.
4. Corrosion resistance.
5. Wear resistance.
6. Cavitation erosion resistance.

■ Description of Rigs, Procedures, Results

1. Maximum load-carrying capacity:

(a) Master-rod bearing test rig – The load-carrying capacities of various bearing materials are determined by means of a test rig which uses a test bearing identical to the crankpin bearing used in production engines. This rig, originally developed by E. A. Ryder, was redesigned to accommodate a master rod bearing used in Cyclone engines. In a radial engine the load on the crankpin bearing varies approximately as the square of the speed, since it is largely due to the centrifugal effect of the connecting-rod assembly. The crankpin bearing load is not very greatly affected by throttle opening, but is somewhat less at full throttle than it is at closed throttle for the same rotative speed. The load travels around inside the bearing instead of being a reversing load, such as occurs on a crankpin serving a single cylinder.

To simulate the above conditions in a test rig, the test bearing is assembled into a bobweight which is carried on a crankpin. This bobweight is kept from rotating with the crankpin by means of a train of gears as shown in Fig. 1. This view shows the rear cover removed, exposing the bobweight-bearing assembly mounted on the crankpin. Fig. 2 shows an exploded view of the bobweight-bearing assembly. Figs. 3 and 4 show the rear and side views of the test rig, respectively. The front end of the crankshaft is supported in the cast-iron housing of the test rig by two ball bearings and is driven by V belts from a variable-speed, 30-hp motor. The crankshaft consists of four major parts, the front crankcheek and crankshaft extension, the rear crankcheek, the counterweight, and the crankpin. The bobweight assembly weighs approximately 72 lb, which duplicates the master balance weight of the Wright Cyclone 9-cyl engine. Since the throw of the crankshaft

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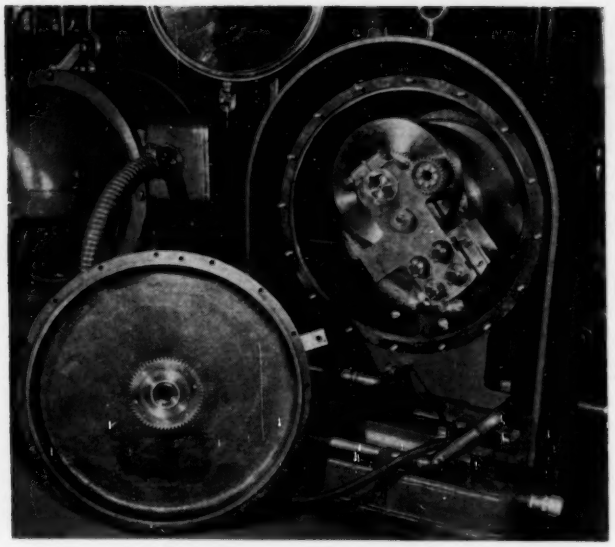
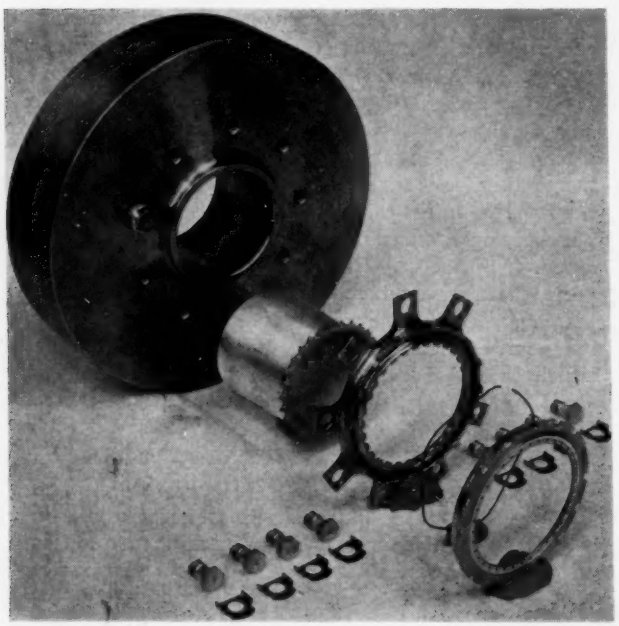
THE AUTHORS: J. PALSULICH (M '36), project engineer with Wright Aeronautical Corp., Paterson, N. J., joined Wright shortly after his graduation from New York University in 1933. Since 1938, he has been in charge of the Bearing Section there which handles all sleeve and anti-friction bearing problems. R. W. BLAIR (J '42) joined Wright Aeronautical Corp. shortly after his graduation from Ohio State University in 1939. Now assistant project engineer, he served as tester and test engineer and was recently named assistant head of the Bearing Section.



■ Fig. 1 – Master rod bearing test rig with cover removed showing crankshaft and bobweight assembly



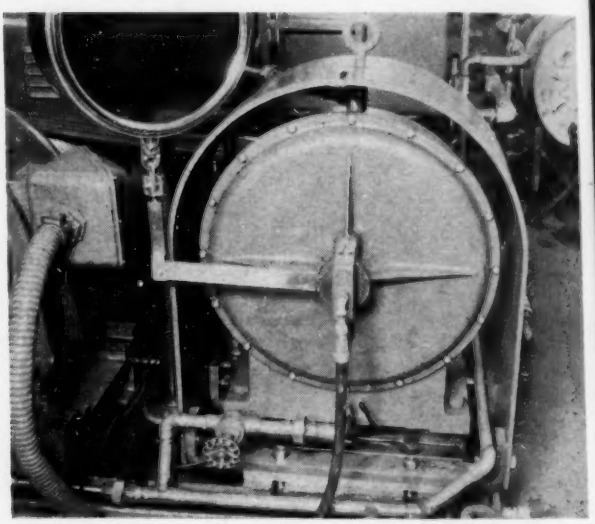
■ Fig. 2 – Exploded view of bobweight assembly used in master rod bearing test rig

is the same as that of the engine, the test bearing loads for any given speeds duplicate those for the full-scale engine operating in the dive condition. The crankpin is removable from both crankcheeks and can readily be replaced as often as necessary.

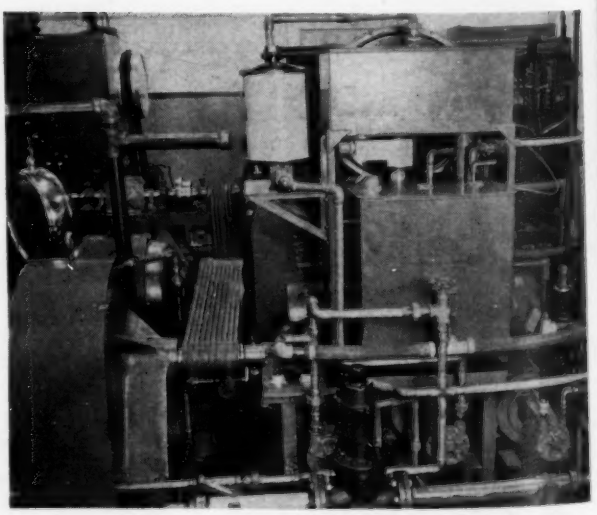
The oil to the test bearing is fed under pressure through a gland on the drive end of the crankshaft. The tachometer drive, an autosyn motor, is taken off the same end. A separate supply of oil is fed to the piniongear bushing through the rear crankcheek and a gland inside the torque arm shaft.

Fig. 5 shows the general arrangement of the test rig, including the motor-generator set and d-c motor, the oil weighing scale, tanks, and ful-flo filter. Fig. 6 shows the instrument panel, including contact-making and recording wattmeters, a recording oil flowmeter and a recording oil-in temperature controller.

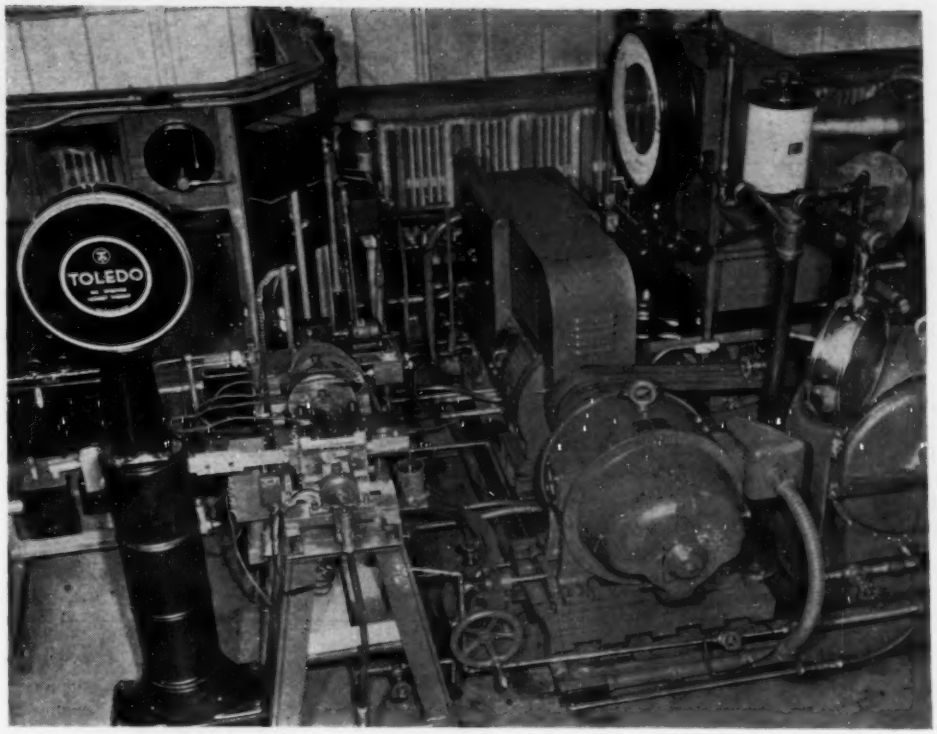
Normally, the test bearing consists of a steel back with a bearing liner, approximately 0.020 in. thick, bonded to the ID. In all cases, the bearings have a loose fit on the OD, are splined at one end and restrained from rotation by mating with the splines on a lockplate. The bearing dimensions are 3.25-in. ID by 3.1 in. in length. The bearing clearance on the ID is 0.0040-0.0055 in. and on the OD it is 0.0010-0.0025 in. When not provided with an electroplated overlay, the bearing surface roughness is 20 microin. maximum. The present crankpins used for these tests are normally AMS 6415 steel nitrided and polish ground to a surface finish of 6 microin. Oil is fed to the bearing through three 0.31-in. diameter holes in the crankpin.



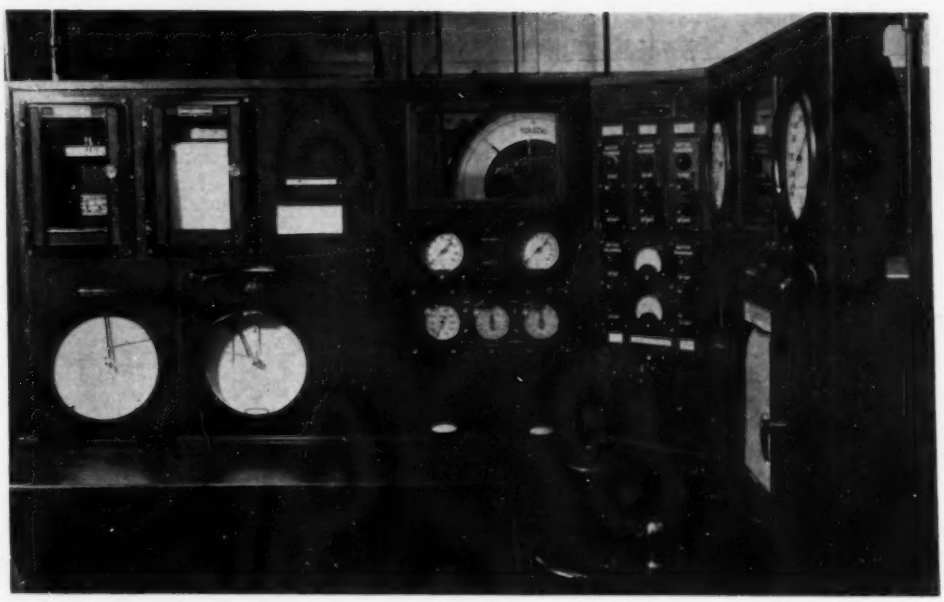
■ Fig. 3 – Master rod bearing test rig completely assembled and ready for test



■ Fig. 4 – Side view of master rod bearing test rig and associated equipment



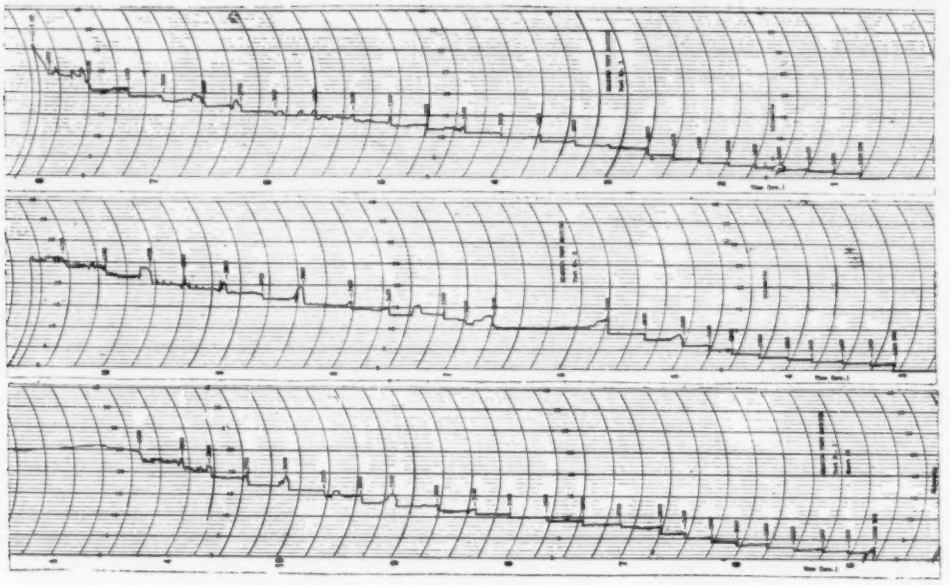
■ Fig. 5 – Rear view of master rod bearing test rig, and four-bearing friction machine with associated equipment



■ Fig. 6 – Control panel for master rod bearing test rig and four-bearing friction machine

The following test procedure is used. Prior to each test, hot oil is pumped through an inlet at the top of the case which directs the oil onto the bobweight. By circulating hot oil for 1 hr before starting the test, the housing and bobweight are brought up to approximately oil temperature so that no difficulties are encountered with bearing clearances due to unequal bobweight and crankpin expansions at the beginning of the test. Oil of 120 SUS viscosity is supplied to the crankshaft at 185 F and 70 psi pressure. The crankshaft speed is increased in 200 rpm increments each 15 min from 1000 to 2600 rpm and in 100 rpm increments each 20 min from 2700 rpm until the end of the test. At each speed, readings are taken of rpm, main oil pressure, pinion bearing oil pressure, oil flow through the test bearing and total flow, oil-in and oil-out temperatures, friction torque, and wattmeter.

Sample wattmeter records for three different tests, using silver, lead-indium plated bearings, are shown in Fig. 7. Comparative load-carrying capacities of various bearing materials are shown in Table 1. It is of interest to mention that good correlation of test results is obtained with those achieved in a dive dummy engine which is used for



■ Fig. 7 – Typical wattmeter charts from the master rod bearing test rig

checking the results of this laboratory test rig.

(b) Dive dummy engine – The results of bearing load-capacity tests on the laboratory test rig, if promising, are checked in a dive dummy engine which is driven by means of a 200-hp dynamometer. This engine is shown in Fig. 8. It is the same as a standard production, Cyclone 9-cyl radial engine except that valves, valve gear, and supercharger are not incorporated. To reduce the cost of testing, a three-piece crankshaft is used instead of the normal two-piece crankshaft, thus permitting frequent crankpin replacement. The bearing and crankpin design, clearances, and surface finishes are identical to those used in the laboratory test rig. Oil of the same viscosity, temperature, and pressure is supplied and the same running schedule is used as in the test rig. Likewise, the load on the bearing varies as the square of the speed and rotates inside the bearing.

Comparative load-carrying capacities of various bearing materials are shown in Table 1.

2. Friction and lubrication characteristics:

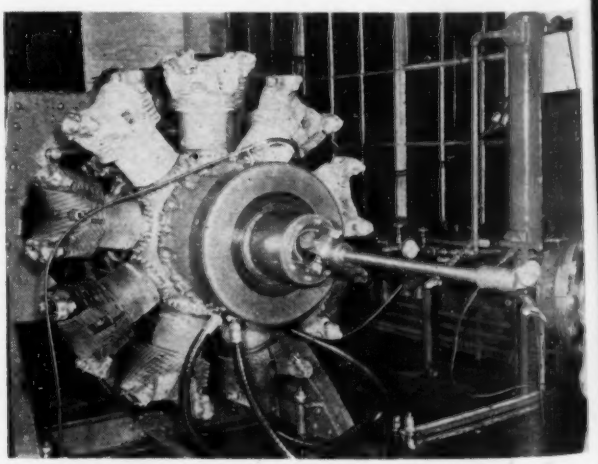
(a) Four-bearing friction machine – The friction and lubrication characteristics of highly loaded sleeve bearings are investigated on a four-bearing friction machine which was designed by the U. S. Bureau of Standards. (See Fig. 9.) Certain changes have been made in the original design which enable higher loads and speeds to be achieved.

The machine consists of four test bearings contained within a housing which floats on the horizontal test shaft and acts similar to a cradle dynamometer. The friction torque is measured at the end of a 20-in. torque arm by means of a scale. The test bearings are mounted in retainer sleeves which in turn are mounted in self-aligning ball bearings. The ball-bearing outer races are pressed into bearing plates. The inner race of each self-aligning ball bearing is prevented from rotation by means of a torque yoke arrangement, as shown in Fig. 10. The two outer bearing plates are rigidly attached to the housing. The two inner bearings are mounted in plates which are un-

Table 1 – Maximum Load-Carrying Capacity of Various Bearing Materials

Bearing Material	Maximum Speed* without Seizure, rpm	Unit Bearing Load, psi	Rubbing Velocity, fps	*PV*
The following tests were run in the master rod bearing test rig:				
A	4000	11,200	57	638,000
B	3900	10,600	55	583,000
C	4200	12,300	60	738,000
D	4000	11,200	57	638,000
E	3300	7,600	47	356,000
F	3100	6,700	44	295,000
G	3100	6,700	44	295,000
The following tests were run in the dive dummy engine:				
A	3900†	10,600	55	583,000
H	3700†	9,600	52	500,000
J	3700†	9,600	52	500,000
K	3600†	9,100	51	465,000
L	3100	6,700	44	295,000
M	3400	8,100	48	389,000
N	2000	2,800	28	78,000

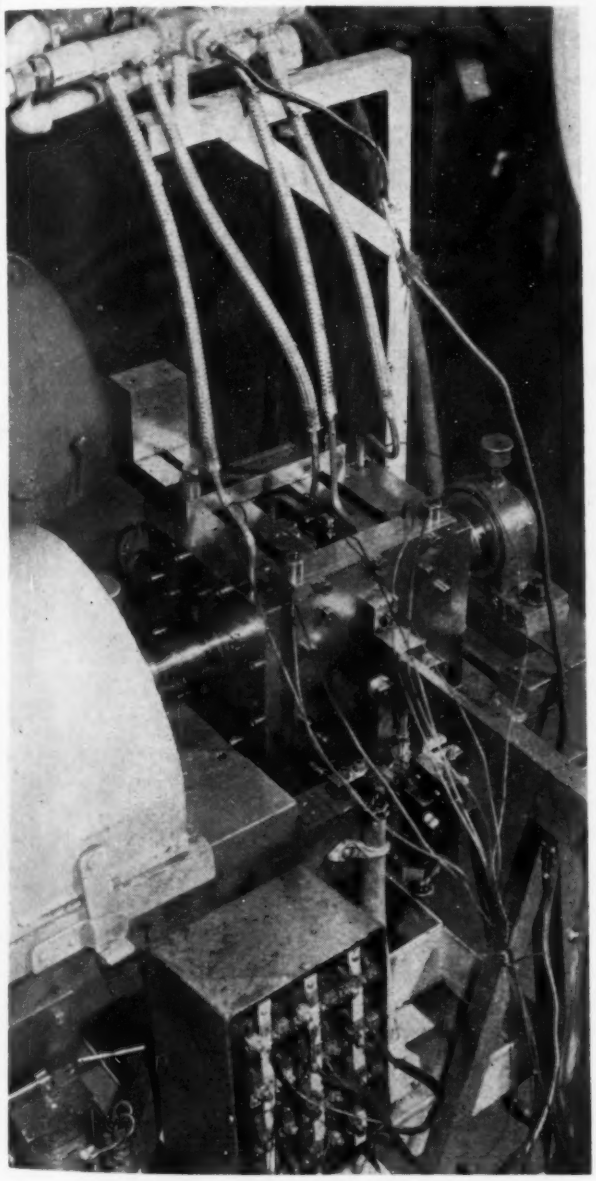
*Seizure occurred at speed 100 rpm above that listed.
†Bearings did not seize. Tests were terminated by equipment limitations.



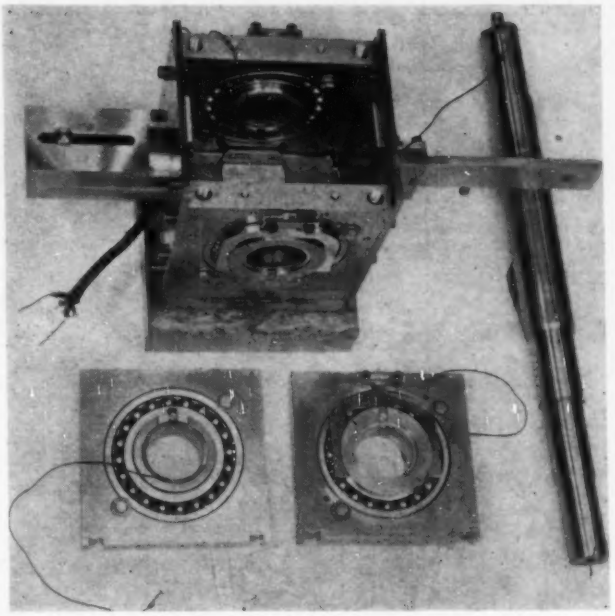
■ Fig. 8 – Wright Cyclone dive dummy engine mounted on test stand

attached to any part of the housing but which slide in vertical guides. Load is applied equally to the two inner bearings from a hydraulic jack at the base of the bearing housing through the two jack plungers. Oil is fed to the hydraulic jack through a coiled tube to prevent interference with torque measurement. The reaction from this load is taken by the two outside bearings and, since the bearings are symmetrically spaced, each will be equally loaded.

An automatic load release is incorporated in the hydraulic loading system. A pin which is attached to the under side of the torque measuring arm contacts a microswitch which in turn actuates a solenoid operated load relief valve. This setup permits the test to be stopped at any predetermined friction torque, thus making possible the inspection of the test bearings at the earliest signs of seizure.



■ Fig. 9 – Four-bearing friction machine ready for test



■ Fig. 10 – Four-bearing friction machine housing assembly with shaft and center bearing trunnions removed

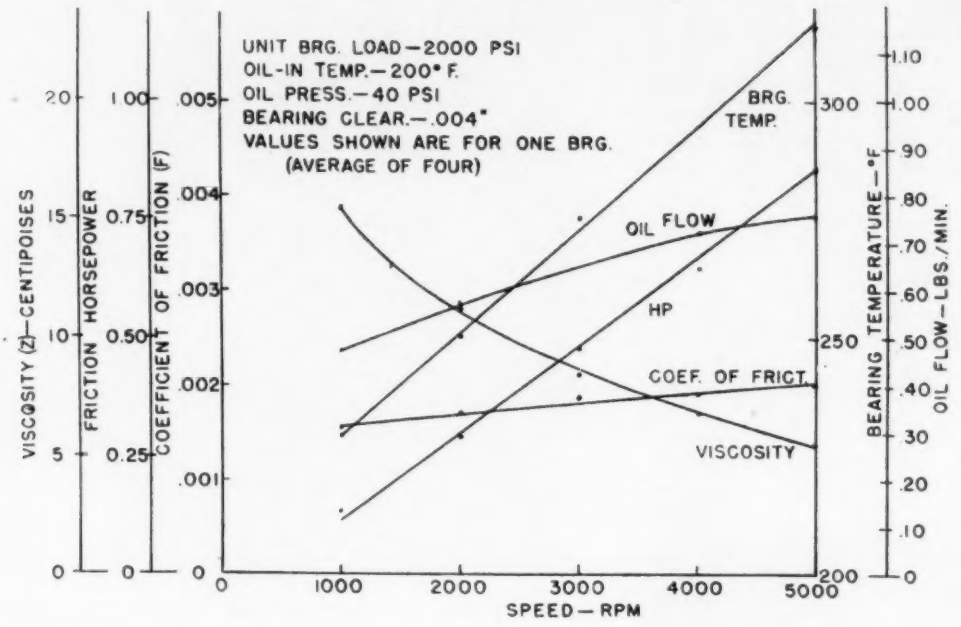
Provision is made for lubrication by means of separate oil lines to each bearing, branching off from a main supply line.

The test bearings are approximately 2 in. in diameter by 1¼ in. long. These are silver-lined steel shells with an overlay of lead-indium on the ID. The test shaft is AMS 6250 steel with a hardness of Rockwell C 60 and finished to 6 microin. Thermocouples are located on each of the four test bearings, on the main oil-in line, and on the oil drain pipe leading from the bearing housing.

Before making the test runs the bearings are subjected to a 2-hr run-in. During tests to determine the frictional and operating characteristics of bearings, the load is held constant and the speed increased from 1000 to 5000 rpm in increments of 1000 rpm. The unit loads used in these runs were 0, 500, 1000, 2000, 3000, and 4000 psi. No attempt was made to operate in the unstable region. The machine is operated at a given load and speed until the bearing thermocouple readings are constant, indicating equilibrium operating conditions. The data are recorded and operation is continued at the next higher speed.

To indicate the nature of the results obtainable from these tests, sample data are shown in Table 2 for oil with a viscosity of 120 SUS at 210 F, and viscosity index of 100. A greater accumulation of this type of data is shown plotted in Figs. 11, 12, 13, and 14. Fig. 11 shows the variation of bearing temperature, oil flow, oil viscosity, coefficient of friction, and friction horsepower with shaft speed, the load being held constant. Fig. 12 shows the same variables plotted against unit bearing load, the speed being held constant. Fig. 13 shows the generalized operating variable ZN/P plotted against the coefficient of friction, while Fig. 14 shows the Sommerfeld variable plotted against the product of diameter-clearance ratio and the coefficient of friction. This type of information is extremely valuable in the design of engine bearings.

3. Fatigue life:

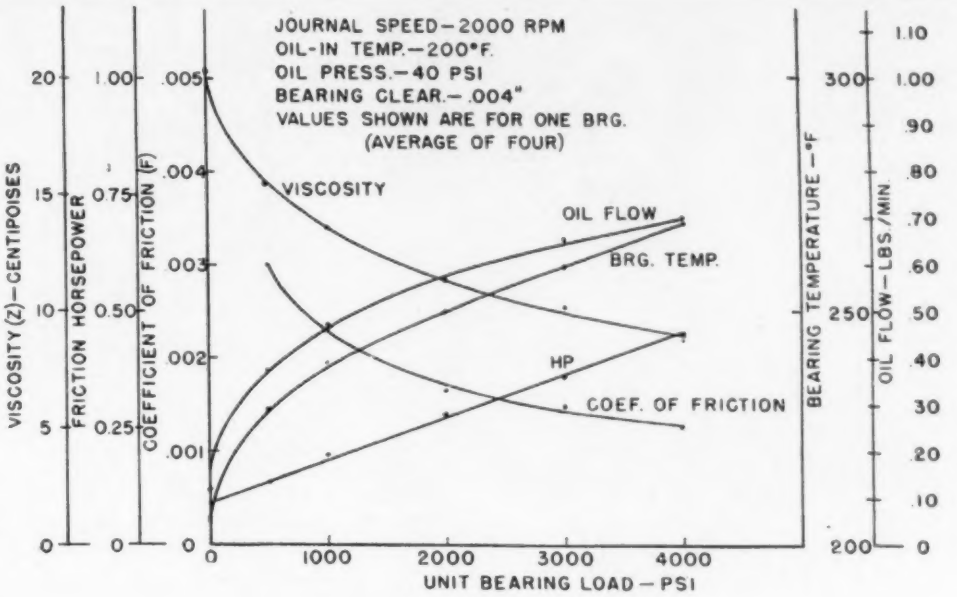


■ Fig. 11—Bearing characteristics curves

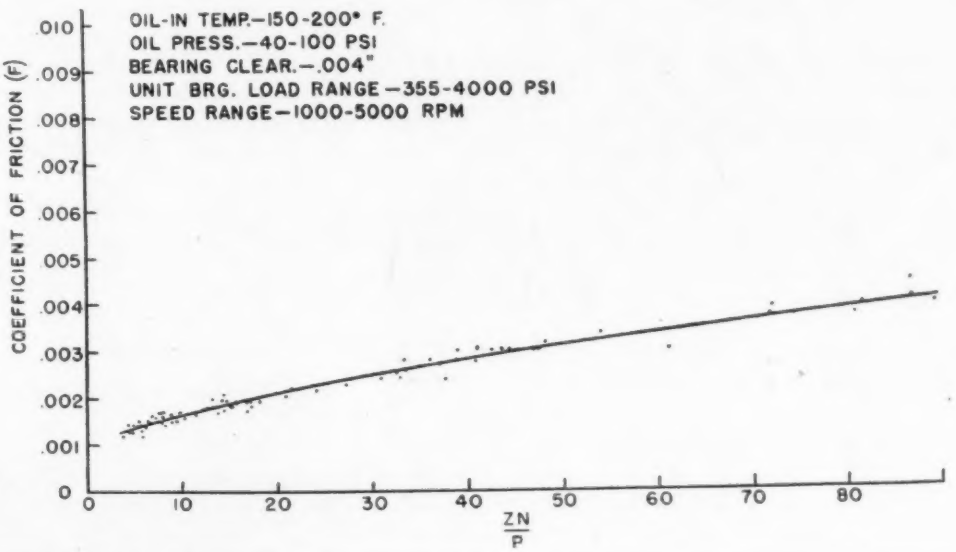
(a) Three-roller fatigue test rig—The fatigue life of various bearing materials and also of various designs of splines on master rod bearings is evaluated on a test rig shown in Fig. 15. This rig provides a method for deflecting a bearing by means of a 1½-in. diameter load roller inserted through the bore of the bearing while supporting the bearing on two power-driven rollers spaced 2 in. apart. A production master rod bearing is used for

Table 2 – Test Results – Four-Bearing Friction Machine

Unit Bearing Load, psi	Rpm	Oil Pressure, psi	Oil-In Temperature, F	Bearing Clearance, in.	Bearing Temperature (Average of Four Bearings), F	Oil Flow per Bearing (Average of Four Bearings), lb per min	Friction Torque per Bearing (Average of Four Bearings), in.-lb	Oil-Out Temperature, F	Case Temperature, F	Room Temperature, F	Friction Hp per Bearing (Average of Four Bearings)	f	Z	ZN/P	$\left(\frac{d}{c}\right)f$	$\left(\frac{d}{c}\right)^2\frac{N}{P}\mu$
							Test No. 1									
0	1003	40	195	0.0040	188	0.0832	2.51	140	100	72	0.0398		46.7			
0	2004	40	199	0.0040	205	0.119	2.76	186	121	73	0.0876		23.0			
0	3002	40	199	0.0040	226	0.137	3.01	211	134	74	0.143		16.4			
0	4006	40	199	0.0040	244	0.155	3.11	227	150	75	0.197		12.35			
0	4988	40	199	0.0040	258	0.178	3.26	239	158	76	0.258		10.2			
							Test No. 2									
500	1027	40	201	0.0040	218	0.270	3.26	209	165		0.0517	0.00241	18.3	37.6	1.24	1.44
500	2002	40	200	0.0040	229	0.287	4.11	220	168		0.131	0.00304	15.3	61.2	1.585	2.35
500	3017	40	204	0.0040	244	0.310	5.02	231	167		0.239	0.00372	12.35	74.5	1.92	2.88
500	3997	40	197	0.0040	258	0.487	5.27	241	171	78	0.334	0.00392	10.2	81.6	2.018	3.13
500	4998	40	204	0.0040	271	0.552	5.52	250	172	79	0.437	0.00409	8.72	87.2	2.11	3.34
							Test No. 3									
1000	1013	40	198	0.0040	222	0.385	5.02	212	174	79	0.0796	0.00186	17.1	17.3	0.958	0.86
1000	1990	40	200	0.0040	238	0.475	6.27	223	171	80	0.199	0.00232	13.5	26.9	1.198	1.03
1000	3016	40	201	0.0040	253	0.520	6.78	233	171	80	0.323	0.00251	10.9	32.8	1.295	1.25
1000	4004	40	200	0.0040	274	0.570	7.54	246	175	82	0.478	0.00279	8.38	33.5	1.44	1.28
1000	4978	40	199	0.0040	287	0.617	7.54	256	180	84	0.598	0.00279	7.28	36.2	1.44	1.39
							Test No. 4									
2000	1006	40	198	0.0040	229	0.470	8.27	214	180	82	0.131	0.00153	15.3	7.69	0.791	0.296
2000	2001	40	202	0.0040	250	0.572	9.04	224	178	83	0.267	0.00167	11.4	11.4	0.881	0.438
2000	3009	40	201	0.0040	275	0.572	10.02	237	181	83	0.477	0.00186	8.3	12.45	0.958	0.478
2000	4000	40	203	0.0040	294	0.715	10.08	256	187	87	0.638	0.00187	6.72	13.44	0.964	0.516
2000	4969	40	200	0.0040	315	0.750	10.81	273	193	87	0.856	0.00200	5.36	13.31	1.03	0.512
							Test No. 5									
3000	1016	40	201	0.0040	225	0.520	10.56	206	131	75	0.167	0.00131	16.5	5.58	0.674	0.214
3000	2000	40	204	0.0040	254	0.655	11.58	219	150	75	0.368	0.00146	10.8	7.2	0.736	0.276
3000	3005	40	204	0.0040	287	0.710	12.06	240	170	77	0.574	0.00149	7.28	7.29	0.768	0.280
3000	3984	40	203	0.0040	309	0.785	12.82	261	189	77	0.813	0.00158	5.7	7.57	0.815	0.291
3000	4962	40	203	0.0040	332	0.872	13.68	276	198	78	1.08	0.00169	4.54	7.52	0.869	0.288
							Test No. 6									
4000	1015	40	200	0.0040	237	0.553	12.54	218	190	78	0.199	0.00116	13.8	3.5	0.598	0.134
4000	2015	40	203	0.0040	270	0.693	13.96	233	187	78	0.444	0.00129	8.87	4.46	0.685	0.171
4000	2985	40	203	0.0040	305	0.770	14.46	256	193	79	0.668	0.00134	5.94	4.43	0.669	0.170
4000	3995	40	203	0.0040	329	0.838	15.03	271	200	80	0.954	0.00139	4.7	4.69	0.717	0.180
4000	4972	40	204	0.0040	370	1.08	15.57	280	208	80	1.23	0.00144	3.33	4.14	0.742	0.198



▪ Fig. 12—Bearing characteristics curves



▪ Fig. 13 – Coefficient of friction plotted against ZN/P

spline strength evaluation, while the same size of bearing without splines is used for bearing material evaluation. In the former tests, recesses are provided in the rollers for spline clearance. Straddling the test bearing is a yoke resting on needle bearings mounted on the ends of the load roller. The load is applied to this roller through a coiled spring in contact with the top of the yoke. The spring is calibrated for loads up to 1000 lb. Pins extending perpendicularly from the end faces of the yoke provide platforms for dial indicator contactors which measure deflection of the bearing under load. The maximum deflec-

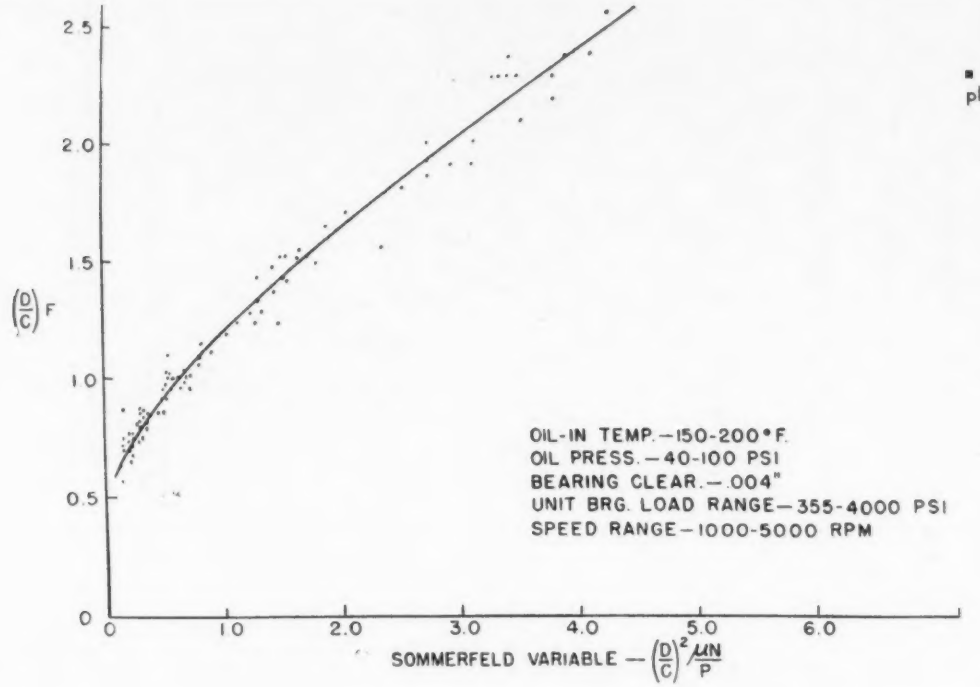
Table 3 – Fatigue Test Results

Material	Load, lb	Running Time before Development of Fatigue Cracks, hr
A	1000	1½
	425	150
B	1000	15
C	425	4
D	425	171½
E	425	125
F	425	500
G	425	1 min
	175	8½
H	425	½
	175	300
J	425	6
K	425	5½
L	425	165

Table 4 – Bearing Corrosion Test Results

	Bearing Material			
	Lead	Lead-Indium (4% Indium)	Lead-Indium (10% Indium)	Codeposited Lead-Tin (10% Tin)
Corrosion Loss, mg per sq cm	73	2.5	0.7	0.2

Note: Thickness of bearing material overlay was 0.003 in. in the case of lead and 0.001 in. in the case of the lead-indium and lead-tin.
The value shown for lead is smaller than the actual corrosion rate, since all of the lead was removed.



■ Fig. 14 – $(D/C)F$ plotted against Sommerfeld variable

tion that has been attained with a steel-backed silver bearing is 0.012 in. The speed of the driving rollers may be varied from 150 to 700 rpm. The rig is driven by a 1½-hp varispeed drive.

The bearing specimen is easily mounted and removed, allowing frequent inspections with little loss in running time. Fig. 16 shows the rig disassembled and a sample test bearing.

The test bearing is partially submerged in oil (SAE 30) while running. Periodic inspections of the bearing are made, varying in running time from 30 min to 1 hr, according to the type of material on test. All tests are run at 700 rpm.

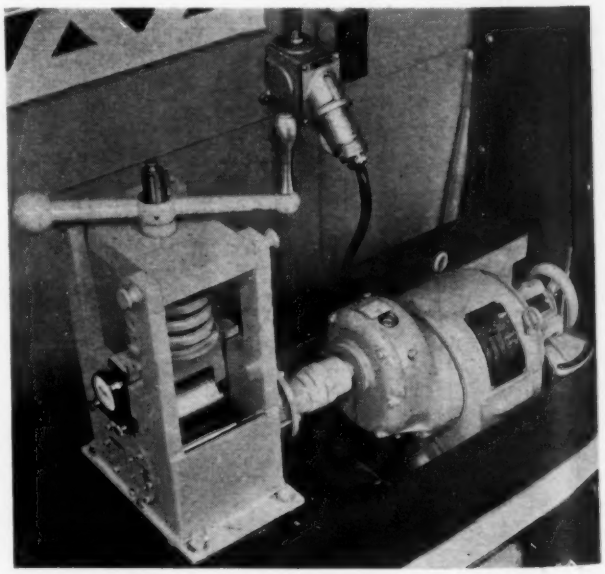
Fatigue test results on various bearing materials are shown in Table 3.

4. Corrosion resistance:

(a) Thrust bearing corrosion machine – For development research, bearing corrosion tests are conducted on a small laboratory machine developed by the Shell Development Co. of Emeryville, Calif. It is known as the tbc Machine and is shown disassembled in Fig. 17. The machine consists of a steel cup filled with oil in which a steel disk is rotated against three flat bearing specimens. Each of the bearing specimens is mounted in a holder which is supported by a ball pivot. These specimens adjust their angle of tilt according to conditions of speed and load and operate as Kingsbury thrust bearing shoes to provide hydrodynamic lubrication. Heat is supplied to the oil by an electric heater set in a recess in the bottom of the cup. The oil temperature is controlled by means of a thermoswitch installed in the steel cup between the oil and heater sections. A graphic temperature recorder has been found very useful, particularly since a portion of the test is run at night with no observer present.

The standard conditions for operation of this test rig are:

Duration of Run, hr	20
Oil Charge, cc	35
Oil Temperature, F	275
Oil Viscosity at 210 F, SUS	120
Oil Neutralization No. (obtained by addition of heptanoic acid)	0.1
Thrust Load, lb	100
Speed, rpm	2500
Brinell Disc Hardness	300
Disc Surface	Mirror finish



■ Fig. 15 – Three-roller fatigue test rig completely assembled and ready for test

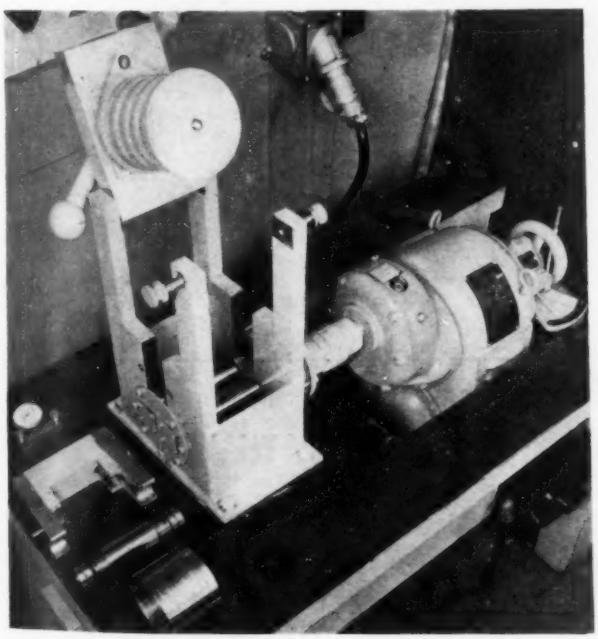
Table 5 – Bearing Wear Test Results – Tabor Abraser

Bearing Material	Condition of Material	Weight Loss g per 200c	Tabor Shear Hardness Under 500-g load
A	As plated	0.121	13
B	As received	0.024	18
C	As received	0.016	24
D	As received	0.009	48
E	As received	0.018	32
F	As received	8.005	83
G	As received	0.003	110
H	As received	0.005	62
			Under 250-g load
A	As plated	0.121	18
J	Diffused for 2 hr at 350 F	0.095	22
K	Diffused for 2 hr at 350 F	0.104	22
L	None	0.062	23
L	Heated for 2 hr in oil at 350 F	0.064	22

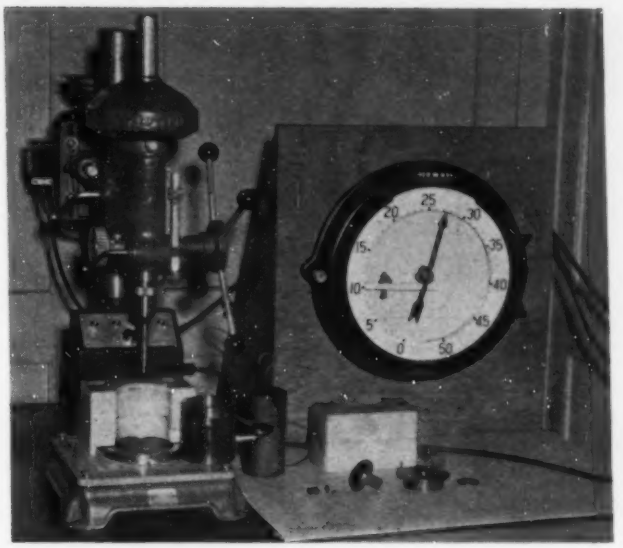
Notes: 1. Type of wheel – Calibase CS-15.
2. Specimen – 4-in. diac, ⅛ in. thick.
3. Overlay thickness (where used) – 0.001 in.
4. Speed – 70 rpm.
5. Load – 500 g.

Bearing Material	As specified
Air Flow into Oil	None

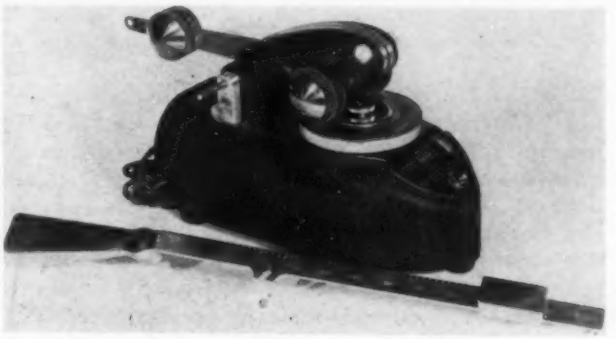
An interesting experience was encountered in the initial test runs on this machine. As originally designed, the shank of the driving spindle was made of steel and the tip was made of fiber. The fiber tips failed due to inferior quality. Tips were then made of aluminum bronze. Highly erratic test results were obtained with these tips. Apparently tip wear against the steel disc produced fine bronze particles which acted as a catalyst, thus greatly exaggerating the normal corrosion weight losses. This condition was remedied by surrounding the steel disc with a micarta cylinder which extends above the oil level and thus prevents the tip wear residue from contaminating the oil. Highly reproducible results were obtained on the few tests conducted with the latest setup.



■ Fig. 16 – Three-roller fatigue test rig disassembled for inspection of test bearing



■ Fig. 17 – Thrust bearing corrosion machine disassembled, showing component parts



■ Fig. 18 – Tabor abraser used for wear testing

Corrosion test results on three different bearing overlay materials are shown in Table 4.

(b) Full-scale engines – To supplement and check laboratory test rig bearing corrosion data, special master rod bearings are prepared and installed in full-scale experimental engines for endurance testing. Only if these results show entirely satisfactory performance will similar bearings be released to production engines.

5. Wear resistance:

(a) Tabor abraser – A real problem in the use of highly loaded bearings in aircraft engines exists in the fairly rapid removal of the overlay normally used on silver and bronze bearings to improve their friction properties. This removal is due to a combination of wear and corrosion.

A simple dry wear test has been devised which yields relative wear rates in the anticipated order and which gives reproducible results. This test is performed on a Tabor abraser (research model) which was originally designed to measure the resistance to rubbing abrasion of a wide range of solid materials and coatings. This rig is shown in Fig. 18. It consists essentially of a turntable which holds and rotates the test specimen and a pair of rubber-base standardized abrasion wheels which produce

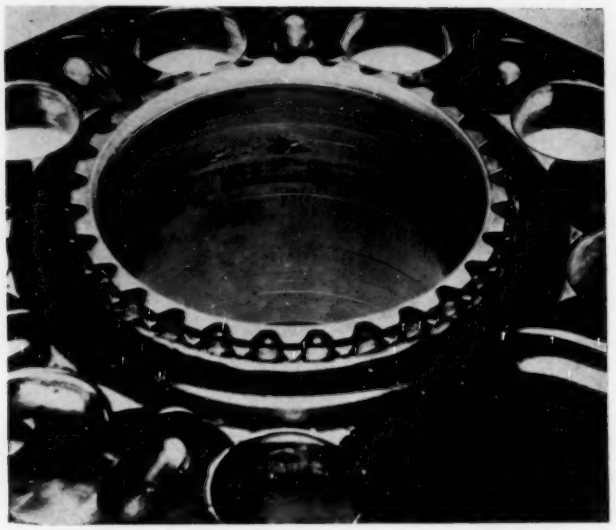


Fig. 19 – Copper-lead bearing (AMS 4820) showing several spots of cavitation-erosion

wearing action on the specimen. These wheels revolve in opposite directions, one sliding radially to the outside, the other sliding to the inside of the wear path. The resulting lines of abrasion are in the form of two arcs crisscrossing each other and the area of the path covers approximately 10 sq cm. The speed of rotation of the specimen is 70 rpm. The load on the abrasion wheels can be varied by means of sliding weights on the load arm. A shear-hardness reading is also taken on the specimens by means of a special attachment. This reading is a measure of the toughness quality of the material.

The most satisfactory test procedure found is as follows:

1. Face the abrasive wheels using the standard facing table and the abrasive discs supplied with the equipment. This operation should be carried out under the test load and should be continued until the faces of the wheels are a uniform green color free from any evidence of loading. This should be accomplished in from 400-600 c. Brush the wheels to remove any loose abrasive material.

2. Degrease, wash with alcohol, dry, and weigh the test specimen to the nearest milligram.

3. Place the weighed specimen on the standard test table and run for 200 c.

4. Remove, clean, dry, and weigh as in (1).

This procedure has given results reproducible in most cases within 5%. Wear test results obtained on this rig are shown in Table 5.

6. Cavitation-erosion resistance:

The phenomenon of cavitation-erosion, frequently encountered in runners of hydraulic turbines, centrifugal pumps, and ship propellers, has been found in a number of highly loaded bearing applications and is a factor to be reckoned with in their design and operation. The action of cavitation results in a physical removal of bearing metal, frequently resulting in holes completely through the bearing liner. (See Fig. 19.) Investigators have propounded many theories on the basic cause of this phenomenon. A logical explanation might be that the attack has its beginning in the formation of cavities within the oil film in the low pressure area following the point of closest approach between the bearing and journal. The subsequent collapse of these cavities in zones of higher pressure can be supposed to result in violent water-hammer effects as the oil penetrates into the empty space created by the cavities. Thus the fundamental characteristic of cavitation is the mechanical water-hammer effect caused by the above-mentioned condition.

To determine accurately the relative resistance of various bearing materials to cavitation-erosion we have set up a program to conduct tests on approximately 35 different materials.

Cavitation-Erosion Discussed Further

The authors' suggestion of cavitation as an explanation for the loss of metal from the surface of bearings was discussed by A. F. Underwood, Research Laboratories Division, General Motors Corp., who has observed that such markings are usually directionally oriented in the direction of a low-pressure area from a high-pressure area. He gave as an example a hole located in a high-pressure area, where the eroded grooves are radially outward from the hole, being deepest at the rim of the hole. He also explained that at the edge of a highly loaded bearing, the eroded grooves deepen as they approach the edge. Very high speed bearings with relatively light loading, he said, will also show loss of metal. These suggest, he continued, the possibility of high-velocity oil causing the erosion just as a hose will wash earth away.

Hydrodynamic bearing theory is based on the axiom that the lubricant velocity is zero at the surface layers, he pointed out, so that if erosion is caused by high-velocity oil, it indicates that the shear forces are greater than can be sustained by the oil-to-metal layer.

The need was expressed by P. G. Exline and W. E. Kramer, Gulf Research & Development Co., for a companion paper to this one in which detailed interpretation of the results obtained in each machine is made in the light of results of such full-scale engine tests as may be available.

The experience of H. W. Luetkemeyer, Cleveland Graphite Bronze Co., with the Tabor abraser has not been as satisfactory as that indicated by the paper, although he explained that he did not believe that his technique was proper. He and his associates are in doubt as to whether abrasive wheel quality is uniform, whether wheel loading is a variable factor during or between runs, or whether rate of loading is a variable depending on circumferential length of the wheel. With these factors not yet determined, he was hesitant to offer criticism concerning this type of test except to say that as yet he was not prepared to admit that wear as produced by this machine is comparable with that experienced in bearings where oil films become such an important factor.

ERRATUM

The combined hydraulic torque converter and coupling referred to as the Brookham in P. M. Heldt's paper on "Transmissions Giving Uninterrupted Acceleration," on pp. 335 and 336 of the SAE Journal *for July, is the Brockhouse. It is manufactured by J. Brockhouse & Co., Ltd., West Bromwich, England.*

SIMPLIFYING THE AIRPLANE FOR THE PRIVATE OWNER

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Chief Engineer, Personal Aircraft Engineering
Consolidated Vultee Aircraft Corp.

THE safe operation of personal airplanes will result, not from a simplified airplane, but from a simplified method of operating an airplane.

Paradoxically, such simplified operation may well be the product of a more complicated mechanism. The development of the automobile is an apt illustration of this point.

The operation of the modern motor car has been greatly simplified by the addition of such complex mechanisms as the automatic choke, automatic transmission, and others.

For this reason, aircraft engineers today must be prepared to deal with the problems of personal aircraft design, not in terms of mechanical simplicity, but in terms of operational simplicity.

The treatment of this subject will seem elementary for the excellent reason that the subject is elementary. The techniques and knowledge required for simplification of the operation of the airplane have been known for many years.

As proof of this, note that tricycle landing gears and stallproof and spinproof airplanes were under development in the early 1930's and that they were flying in 1937.

At that time no regulations existed under which such aircraft might be licensed. Yet under special waivers requested of the CAA by their manufacturers, type certificates were granted on four stallproof, spinproof airplanes with tricycle landing gear, namely, Steerman Hammond, TC 644, June 24, 1937; Gwinn Aircar, TC 682, May 27, 1938; Ercoupe, TC 718, March 25, 1940; General Skyfarer, TC 742, Oct. 22, 1941. It is apparent that the techniques for producing safer airplanes have been known for some time. The responsibility for not using these techniques in production sooner rests with the aviation business itself.

One obstacle preventing earlier acceptance of these aids to safer flying was the now dying tradition of heroism in aviation. Another has been the tradition of useless precision in flight maneuvers. In the first days of aviation, it is true that precision was a necessity since high speed was not too much above stalling speed, and a small lack of precision in a turn could produce a stall or spin and a crash. But that period is definitely ended.

Perhaps the greatest resistance to simplified control, however, was found among the distributors of airplanes. They made their profits, not out of the distribution of airplanes, but out of charter services and training operations.

Government regulations have required that pilots have training in spins. The schools, therefore, could not use stallproof, spinproof airplanes for training, and their profits from the sale of airplanes were too low, compared to the profits of training, to warrant their handling one kind of airplane for sale to owners and another for training. Fortunately, this condition is changing.

A typical attitude of 1938 is revealed by Fig. 1. A questionnaire was sent to over 100 airplane distributors in 1938. The sample answer shown, received from one of today's most ardent advocates of stallproof, spinproof airplanes, reflects the general tone of the answers received.

[This paper was presented at the SAE National Aeronautic Meeting, New York, April 4, 1946.]

SAFETY in private flying will result, the author claims, not from simplified airplanes but from a simplified method of operating them.

The author offers as a solution to this problem of operational simplicity a general specification for private airplanes such that the operation will make maximum use of the conditioned reflexes of an automobile driver.

The problem resolves itself, according to the author, into the following factors:

1. **Elimination of need for coordination.**
2. **Elimination of need for depth perception.**
3. **Elimination of nose-down contact with ground.**
4. **Reduction of minimum cruising speed to decrease need for visibility.**
5. **Reduction of landing speed.**
6. **Reduction of temptation to carelessness.**

Let us cease trying to mold people to fit the airplane, the author pleads, and let us fit the airplane to people as they are.

THE AUTHOR: J. M. GWINN, JR., (M '45) has since 1920 been engaged in aeronautical engineering, design, and research. Except for three years with the Gallaudet Aircraft Corp., he has, since serving with the USAAF in World War I, been associated with Consolidated Vultee Aircraft Corp. While on leave from Consolidated from 1935 to 1940, he designed and built the Gwinn Aircar, a stallproof, spinproof airplane. Mr. Gwinn is now chief engineer of the personal aircraft section of Consolidated.

(1). Are you interested in buying tricycle gear light airplanes? Yes

(a) To operate as instruction ships? Yes ☐ No ☒
(b) To sell to customer? Yes ☐ No ☒
(c) To carry passengers for hire Yes ☐ No ☒

(2). Are you interested in buying "stall proof-spin proof" airplanes? Yes

(a) To operate as an instruction ship? Yes ☐ No ☒
(b) To sell to private owner Yes ☐ No ☒
(c) To carry passengers for hire Yes ☐ No ☒
(d) With conventional controls Yes ☐ No ☒
(e) With two controls (no rudder) Yes ☐ No ☒

(3). Do you favor tandem or side-by-side seating

(a) For instruction purposes? Tandem ☐ Side-by-side ☐ ?
(b) For sale to private owners Tandem ☐ Side-by-side ☐

(4) Are you interested in a 3-place light airplane? Yes ☐ No ☒
If answer is yes, do you favor

(a) One seat forward and 2 seats side-by-side, aft? Yes ☐ No ☐
(b) Two seats forward side-by-side and one seat aft? Yes ☐ No ☐
(c) Seats as in (a) for carrying passengers, convertible to 2-place tandem for training. Yes ☒ No ☒

(5). What power do you prefer for

(a) Instruction? 125 hp 150 hp wheel
(b) Sale to private owner?
(c) Use in 3-place light plane?

(6). Do you prefer stick or wheel for training

(a) On conventional airplanes
(1) with tandem seating Stick X Wheel
(2) with side-by-side seating Stick Wheel either
(b) On tricycle gear airplane Stick Wheel
(c) On 2-control airplane Stick Wheel

(7). Do you prefer stick or wheel for sale to private owners

(a) On conventional airplanes
(1) With tandem seating ~~Stick~~ Wheel
(2) With side-by-side seating Stick Wheel
(b) On tricycle gear airplane Stick Wheel
(c) On 2-control airplane Stick Wheel

(8). Would you like flaps

(a) On training airplane? Yes ☐ No ☒
(b) On airplane for sale to private owner? Yes ☐ No ☐
(c) Tricycle gear airplane? Yes ☐ No ☐

■ Fig. 1 – Typical questionnaire on 2-control aircraft in 1938

It is quite evident that in 1938 manufacturers proposing to sell airplanes having simplified control could have made no headway. This situation continued throughout the war, inasmuch as the CPT program was directed toward training of military pilots who were required to know how to spin.

Misdirected activity by the CAA toward reducing hazards phychologically important, but statistically actually not important has been another source of delay in the development of safer aircraft.

In Fig. 2 are shown accidents due to stalls and spins. Note that whereas a large percentage of fatal accidents has always been due to stalls and spins, the activity of the industry devoted to eliminating this hazard, as measured by the percentage of CAA regulations dealing with this hazard, also shown on Fig. 2, has been, until recently, very small. The increase in regulations in 1945 is due to issuance by CAA in November, 1945, of CAR 03, which includes recognition and regulation of stallproof, spinproof airplanes.

Fig. 3 shows the accident rate due to structural failure and the amount of CAA regulations pertaining to structure. Note that whereas structural failure has been for a long time a minor source of hazard, 70% of our activity has been directed to reducing this hazard from 10% to 5%, as measured by the 50% of CAA regulations on structure, and another 20% on detail design and construction.

In Fig. 4 are shown accidents due to powerplant failure, and CAA powerplant regulations. Note again that although this hazard is not increasing, and is, in reality, secondary, CAA has recently found reasons to triple regulations on this subject.

The thesis of this paper is that the principal hazards in flying are, first, the necessity for skill and for judgment; and second, the frequency of catastrophe that follows the use of poor technique or bad judgment. It will also be shown that this frequency of catastrophe is related to minimum controllable flying speed, and the inherent tendency of an airplane to strike the ground out of control with the nose down.

The principal difficulty and the one which makes skill a requirement for safe flying is the six degrees of freedom of motion obtainable in an airplane. These motions are: translation vertically, laterally, and longitudinally, and rotation about the vertical, transverse, and longitudinal axes. By comparison, an automobile, as far as control is concerned, has but two degrees of freedom, the other four motions being restrained by the physical nature of the highway.

The major hazard with six motions is that the airplane may move in a direction other than that which it points. When an automobile has such a motion, it is in a skid, which is a type motion greatly increasing the difficulty of control.

Not only can an airplane move *laterally* in a direction not pointed, but likewise it does not necessarily move in *elevation* in the direction that it points. With its nose up slightly, the airplane may be either climbing, gliding, or flying level.

In gliding to an airport the nose may be down, level, or up. The glide angle of the airplane relative to the air is difficult to judge. This angle is steeper at both very low speeds and very high speeds, and flattest at an intermediate speed. This fools the pilot in that he may think that if he is undershooting the field, lifting the nose of the airplane will make him reach the field. The contrary is correct. This problem is made more difficult by wind, which changes the glide angle at all airplane speeds.

Added to the general problem of six degrees of freedom are the complexities of the control system used to control these six motions. This control system comprises four controls, namely, elevator, aileron, rudder, and throttle.

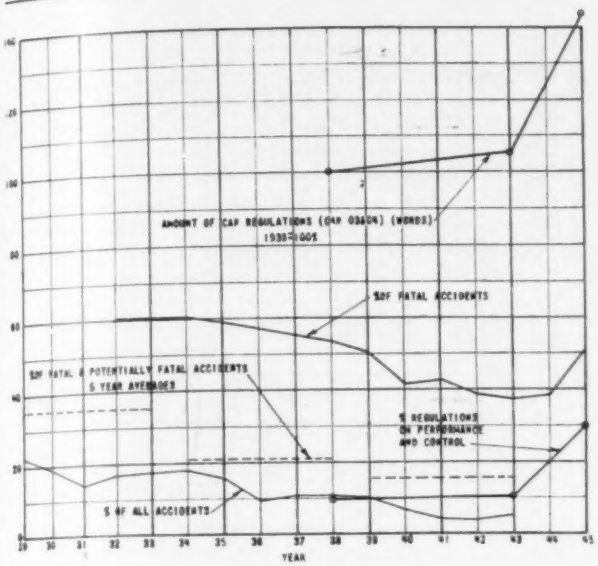
The elevator controls the angle of attack, and therefore only the speed. It does not permanently control the attitude of the airplane relative to the horizon. This attitude is a function not only of speed but of power. The elevator controls, indirectly, rotation in pitch, in that moving the elevator produces rotation in pitch until the airplane has reached equilibrium about the new trimming angle of attack. A secondary effect is a change in altitude, the change in speed producing a change in the kinetic energy, which changes the potential energy and, therefore, the altitude.

Steady vertical motion, however, is controlled by power. There are secondary effects of power which affect trim and, therefore, speed. Among these are moments of thrusts and the effects of slipstream on control surfaces. Fundamentally, however, climb at constant air speed is solely a function of power.

The rudder controls the steady angle of yaw and controls rate of turn only temporarily while the yaw is changing from one angle to another. The actual turning of the flight path in space (as contrasted to merely yawing the airplane) is accomplished by banking, which produces a horizontal component of the lift to push the airplane sideways.

Although banking is produced by aileron, actually the aileron controls rate of roll, banking being an equilibrium result of many factors.

The pilot, therefore, is faced with the problem of controlling six motions with four controls, only two of which independently control one of the six motions. This requires that pilots be trained in coordination.



■ Fig. 2 – Data on accidents due to stalls and spins

Errors in technique and judgment are responsible for approximately 51% of all accidents (42% technique, 9% judgment), and I believe that it may be assumed that most errors of technique are errors of coordination.

Examples of the necessity for and complexity of the coordination problem are shown in Figs. 5, 6, and 7. Fig. 5 shows the relation of aileron, rudder, and elevator movements in a turn of an O3U-1 airplane. Note the variations both of the amplitude ratios and of phase relationships of rudder and aileron deflections.

The turn is started with both right aileron and right rudder, the rudder deflection being several times the aileron deflection. Then the aileron is strongly reversed to prevent overbanking, followed out of phase by a very slight reversal of rudder.

In coming out of the turn, rudder is held neutral, with opposite aileron, and the turn is ended with right rudder and zero aileron. Possibly engine torque and slipstream effects required right rudder for level flight, and the right rudder position could be considered neutral for the rudder.

Fig. 6 shows a vertical turn in the F6C-4 airplane. In this case the amplitude and phase relation of aileron and rudder are together for the first 3 sec going into the turn, but change radically in the turn and in coming out of the turn.

The general trend is the same as in Fig. 5, with strong opposite aileron and slight opposite rudder in the steady turn, and neutral rudder and some opposite aileron coming out of the turn.

Fig. 7 shows the same pattern, even though for a twin-engined airplane. It is probably valid to assume that the effect of twin engines is to increase damping in yaw, due to the outer propeller developing less thrust in a turn (higher air speed at constant power).

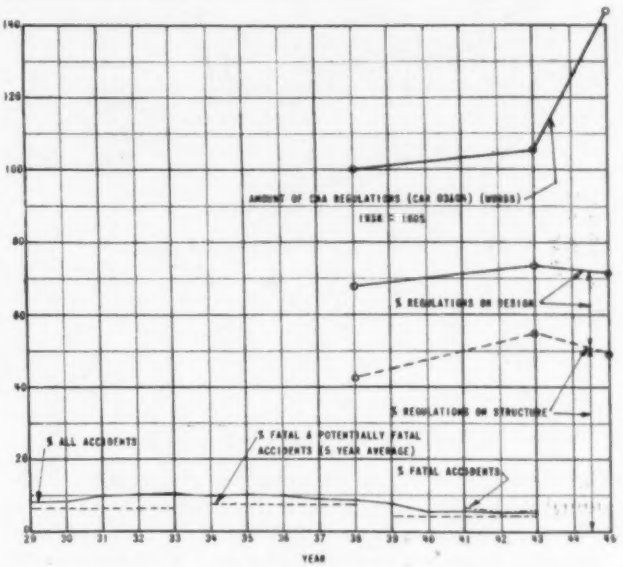
On the ground, the pilot has another set of control problems. Here he is faced with three degrees of freedom, namely, speed, turning, and pitching, and the functioning of his four controls differs on the ground from their functions in the air.

For example, on the ground the throttle and brakes are speed controls, not the elevator. The elevator is solely a pitch control, although the brakes are also a pitch control, and frequently produce nose-over.

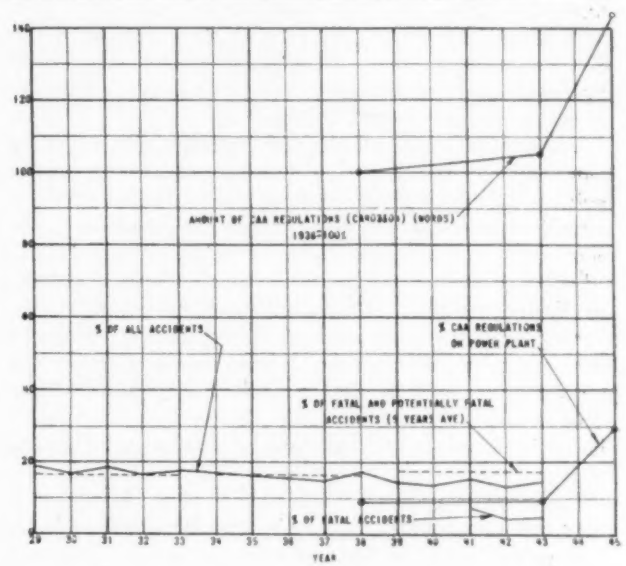
Turning on the ground may be made either by rudder, by differentially operated brakes, or by steerable nose or tail wheel. When turns are made by rudder or brakes, a turning moment is applied, which is not solely a function of the control position but is also a function of the wind direction and velocity, the condition of the terrain, and the tread of the landing gear.

The path resulting from this variable force is likewise variable. The steerable nose or tail wheel, on the other hand, gives direct path control with the radius of turn being a direct function of control position.

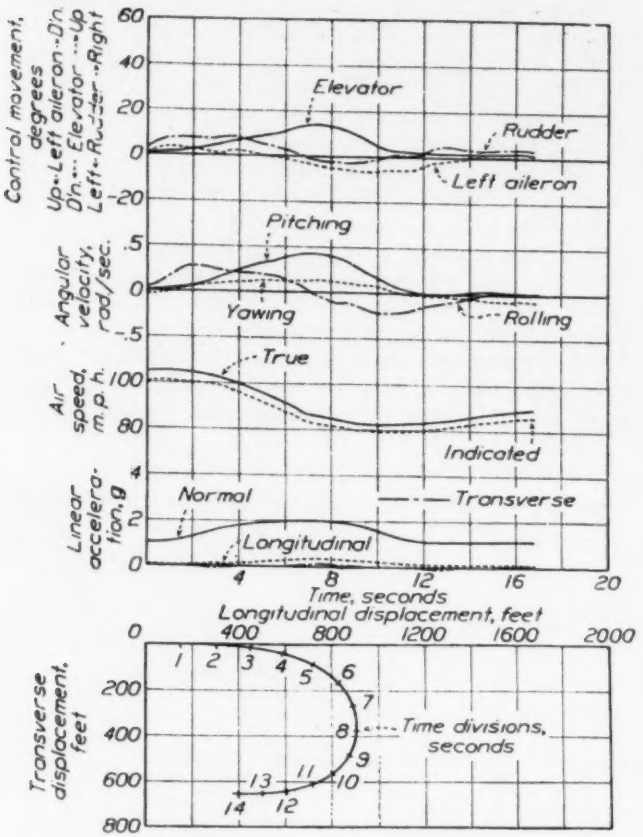
Fig. 8 shows that, while accidents per airplane have not decreased greatly in 15 years, the fatal accidents per airplane have been reduced by three-fourths. Plotted also in


AMOUNT OF CAA REGULATIONS (CAR 03404) (WORDS)
1938 = 100%
% REGULATIONS ON DESIGN
% REGULATIONS ON STRUCTURE
% ALL ACCIDENTS
% FATAL & POTENTIALLY FATAL ACCIDENTS (5 YEAR AVERAGE)
% FATAL ACCIDENTS
YEAR

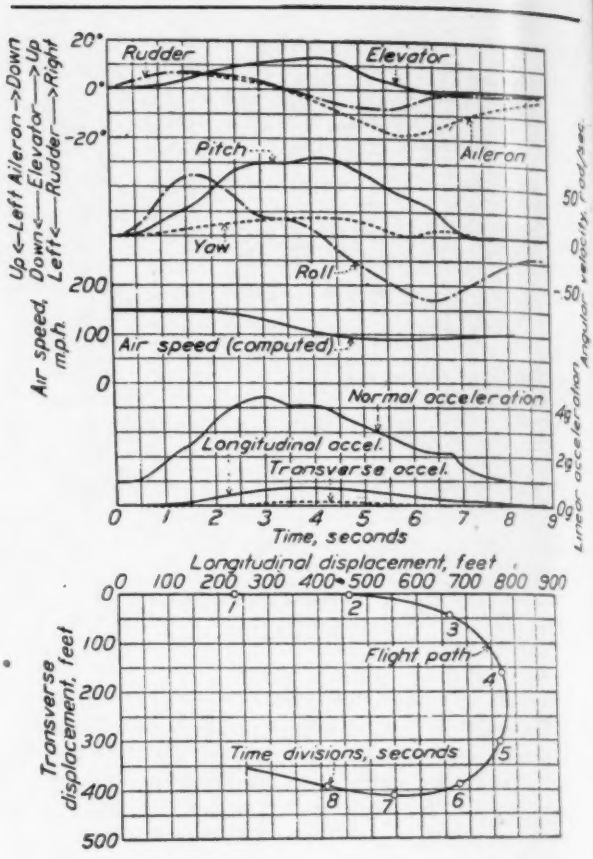
■ Fig. 3 – Data on accidents due to structural failure


AMOUNT OF CAA REGULATIONS (CAR03404) (WORDS)
1938=100%
% OF ALL ACCIDENTS
% CAR REGULATIONS ON POWER PLANT
% OF FATAL AND POTENTIALLY FATAL ACCIDENTS (5 YEARS AVE.)
% OF FATAL ACCIDENTS
YEAR

■ Fig. 4 – Data on accidents due to powerplant failure



■ Fig. 5 – Control movements in turn of 03U-1 airplane



■ Fig. 6 – Control movements in turn of F6C-4 airplane

Fig. 8 are gallons of oil per mile, and percentage of total annual sales that are in the light plane category.

Keeping in mind that the light planes universally have horizontally opposed engines which have very low oil consumption, these two latter curves show that the percentage of airplanes in the light plane category using horizontally opposed engines has been continuously increasing. The light planes generally have much lower landing speeds than other private owner airplanes, and while lower landing speed does not decrease number of accidents, it greatly decreases the damage and fatality ratio.

The airplane itself provides some hazards over which the pilot has no control. While they are not a part of the problem of making the airplane easier to fly, they are certainly a problem in making it safe to fly and should be mentioned.

In Fig. 9 are shown accidents by kind. It is evident from this figure that fire in the air and structural failure are the two principal types of accidents which can be caused by malfunction of the airplane. Forced landings themselves usually are caused by airplane failure (engine), but the accidents due to forced landings are partly a function of the skill and judgment exercised in executing the forced landing.

Wind and weather are a further source of hazard in flight. In cross-country flight, wind produces drift and, the airplane not flying in the direction pointed, misleads the pilot. In landing, cross wind produces drift, and any wind changes the glide angle – again misleading the pilot.

The principal weather hazard is visibility. For contact flying, the *required visibility* is a *function* of *minimum controllable flying speed*. For cross-country navigation by radio this is, of course, not true.

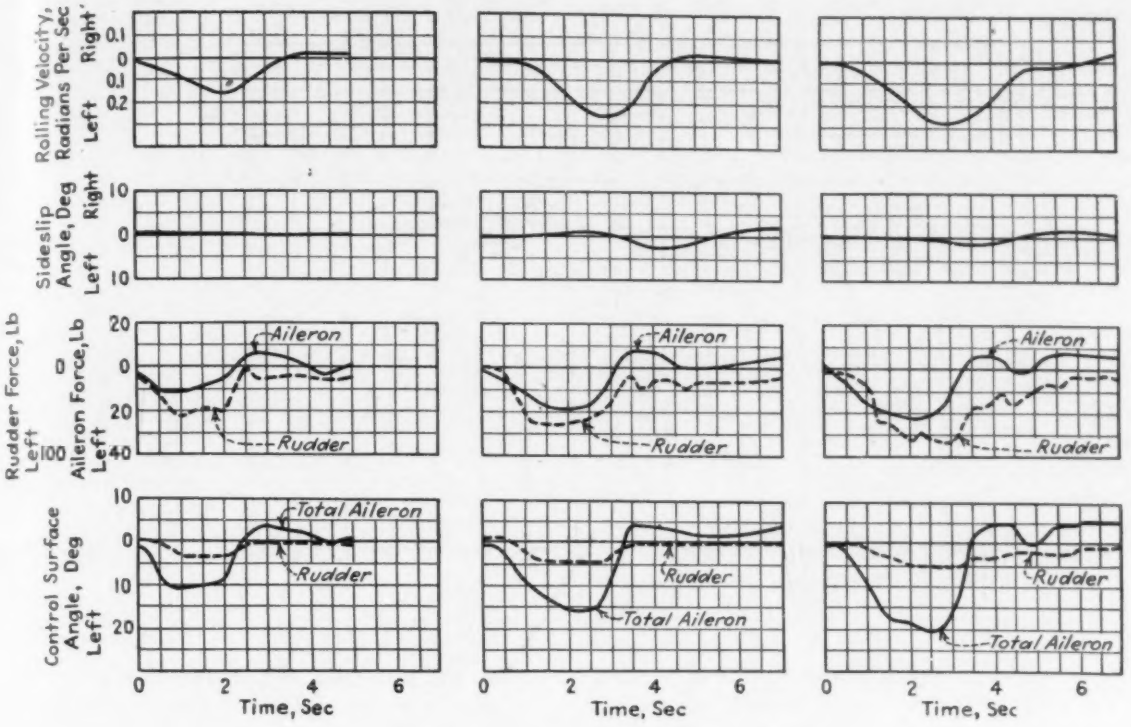
Added to the above difficulties of an airplane is an even more serious drawback. The penalty for error is frequently terrible. This is due first to the speed of motion, which may be sufficiently great so that collision with the ground or some other obstacle occurs before a recognized error can be corrected (particularly true in landing and take-off because the altitude necessarily is low), and secondly, to the fact that, in colliding with the ground out of control, the airplane is usually not level but nose down.

Nose-down crashes are very damaging both to aircraft and personnel. The CAA regulations recognize this and require that safety belts have a factor of 9 longitudinally, compared to 3 upward and 1½ sideways. On the other hand, a level crash can be quite safe for personnel, who can stand 20 to 30 g's vertically in a sitting position.[1]

Collisions in flight, while not a great source of accidents, produce a large percentage of fatalities. Statistics show that over the 15-year period 1929 to 1943, inclusive, collisions with other aircraft were about 10% of all collisions, and collisions with objects other than aircraft, 90%.

During the last four years of the 15, the ratios were about 15% and 85%, the prior 11 years averaged 6% and 94%. The potentially fatal accident ratio is about the same for both collisions with objects and collisions with other aircraft.

[1] National Research Council Committee on Aviation Medicine, Fifth Crash Injuries Conference, Washington, D. C., Oct. 30-31, 1945.



■ Fig. 7 – Control movements in turn of twin-engined airplane

The CAA analyses give carelessness as the major cause of collisions with other aircraft, and carelessness and bad judgment as the major causes of collisions with other objects.

One might assume that visibility was a major cause of collisions. However, since collisions with other objects are nine times as frequent as collisions with other airplanes, which other objects must always have been in the flight path of the airplane, and since vision forward into the flight path, except when the nose is up in landing, is usually pretty good, it is evident that lack of ability to see was not the major factor, at least in 90% of collisions.

This does not mean that we should ignore vision forward and down. It is particularly important in taxiing and navigation, but we must not be trapped into the conclusion that airplanes should be flying green houses.

With the difficulties thus reviewed, we may approach the problem along five lines. We must: (1) reduce the necessity for coordination and skillful technique, (2) make it more difficult to be careless, (3) reduce the hazards from faulty judgment, (4) reduce hazards from low visibility, and (5) reduce the probability of catastrophe when accidents do happen.

In applying these principles to the design of an airplane, it is necessary first to determine their order of importance. Taking the years 1941, 1942, and 1943, the last three years on which complete data are on hand, we find that of fatal accidents, 50.8% are caused by control problems (stalls, spins, landings, take-off, and taxiing), 25% by collision in flight, 6.6% by structural failure, 6.1% by forced landings, and 1.5% by fire.

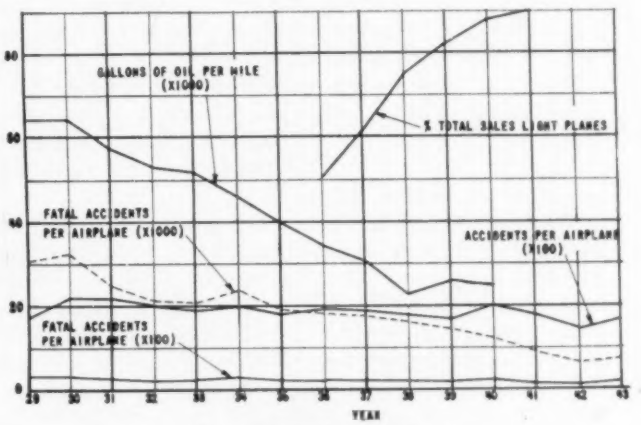
Based on washouts of airplanes rather than fatalities, these statistics become 50.8% for control, 17.7% for forced landings, 17.5% for collisions, 4.7% for structure, and 2.6% for fire.

If we take an average between fatalities and washouts, we obtain control 50.8%, collision 21.2%, forced landings 11.9%, structure 5.6%, and fire 2.1%.

The CAA in its accident analyses also apportions responsibility for each accident on causes. The above given division of accidents is not entirely on the basis of causes, but on the type of accident.

Referring directly to the CAA analyses by causes, we find that for the five years 1939 to 1943, accident causes were in the following percentages: poor technique 41.7, carelessness 16.7, errors in judgment 9.3, weather and darkness 4.5.

This, then, gives us their order of importance, which


GALLONS OF OIL PER MILE (X1000)
$ TOTAL SALES LIGHT PLANES
FATAL ACCIDENTS PER AIRPLANE (X1000)
ACCIDENTS PER AIRPLANE (X100)
FATAL ACCIDENTS PER AIRPLANE (X100)
YEAR

■ Fig. 8 – Annual accident rate

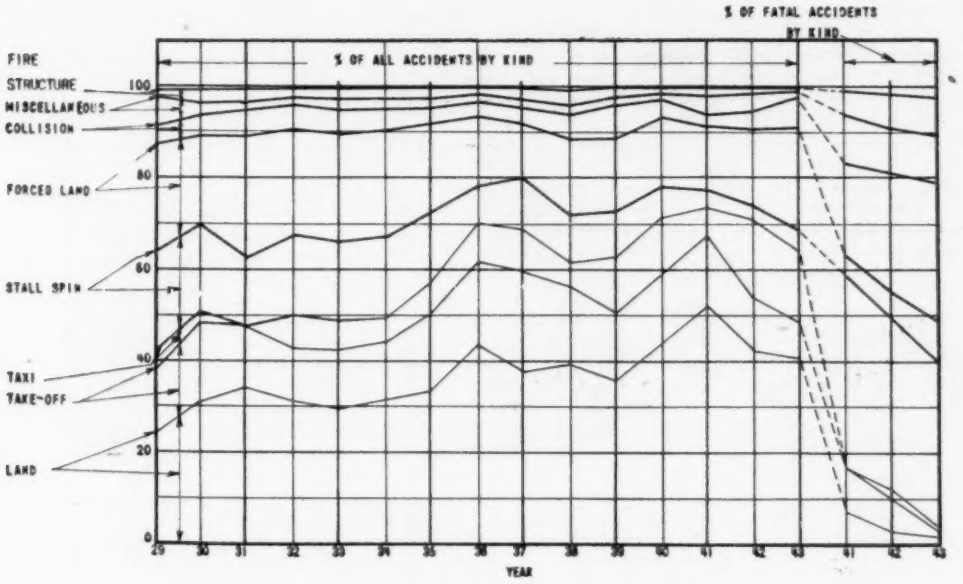


Fig. 9 – Distribution of accidents by kind

conforms to the order previously stated. It is to be noted that as the utility of the airplane is increased and the private owner does more and more cross-country flying, the hazards from weather will increase.

The solution of the control problem requires a reduction in the complexity of control by eliminating unnecessary combinations of motions. The simplest conception is that of an airplane which always goes in the direction pointed, both vertically and laterally. This produces three degrees of freedom – turning, change in elevation, and change in speed, but one more motion than that of an automobile.

This conception is quite satisfactory in the air, but creates difficulties in landings and take-off. Even though in still air the airplane glides in the direction pointed, in a wind it does not glide in the direction pointed. The pilot, therefore, is not saved the problem of developing judgment of glide in winds of various velocities. Furthermore, in landing, due to cross wind, the airplane does not point in the direction traveled relative to the ground.

Assuming for the moment that the pilot can develop sufficiently good judgment of glide, he will collide with the ground nose downward if he does not develop depth perception. This is the most hazardous type of accident, because it produces deceleration forces parallel to the longitudinal axis of an airplane.

Neither the airplane structure nor the occupants are well suited for forces in this direction. The damage to people is high because they are thrown forward against whatever structure is ahead of them. This type of accident produces a high rate of personal injury.

The conception of an airplane that goes in the direction pointed will, therefore, be modified to one which goes in the direction pointed laterally, but is substantially level longitudinally at all times, or one in which, if exact longitudinal level cannot be achieved, deviations shall be in the direction of nose up, not nose down.

The pilot must still learn to judge glide, but at least he will not be fooled by the thought that he is gliding toward the point on the ground toward which the airplane is pointing.

This will then bring home to him that the throttle is still the altitude control, and he will come in with part throttle and use throttle for control of his glide angle. No added glide control need be, nor should be, added. If it is, control is made more complex, not simpler, and the real problem is not to control the glide angle but to *know in time whether the angle of glide* is wrong.

A major advantage of keeping the airplane level, including during the glide, is that it can be flown into the ground without necessity for flare. The airplane structurally should be designed for this type of landing, and if so designed will eliminate the necessity for depth perception.

Naturally as pilots become more skilled, they will make a slight flare before landing to reduce landing loads on their airplanes. But by making flares unnecessary, a great deal will be accomplished to eliminate landing accidents both by reducing skill and judgment required in landing and by eliminating nose-down contacts with the ground.

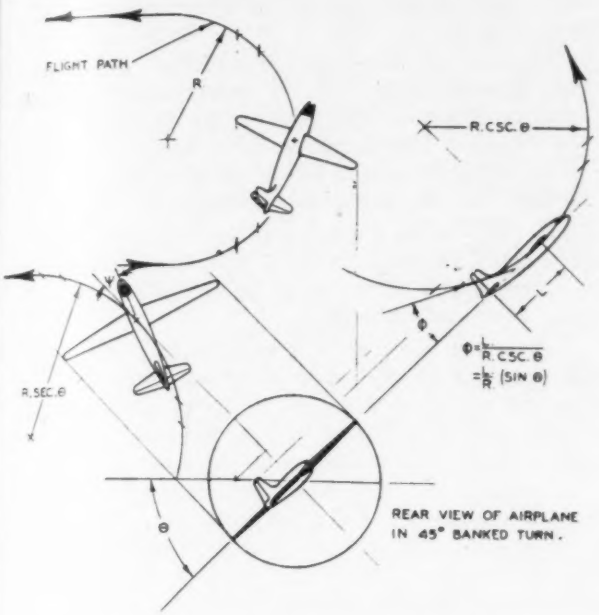
The problem of drift in cross-wind landings can be approached from either of the following two standpoints:

1. Require the pilot to detect it and give him means for correcting it just prior to landing.

2. Make an airplane which automatically corrects its heading on landing and which is capable of taking the loads generated by this action.

To solve the visibility problem, an airplane is required which can fly continuously at a reasonably low speed with complete control and with satisfactory engine cooling. With present techniques, 50 mph should satisfy this requirement, although 45 would be much better. An airplane with this low a flying speed can fly contact over known terrain in rather bad visibility. Furthermore, it will permit approaching landing areas at low altitude and low speed. With such an approach to the edge of the airport, the airplane can be landed by closing the throttle and permitting the airplane to settle on the airport in a horizontal position without change of air speed.

The visibility problem will certainly be increasingly solved by the use of radio and radar, but there will always be a need for contact flight in poor visibility. The visi-



▪ Fig. 10 – Constant altitude turn with inward slip

bility hazard cannot be controlled by regulations, because regulations of this type are not enforceable. As private flying grows, the policing problem alone will be impossible. People resent and disobey laws of the type which attempt to protect them from themselves.

Even those who do not begin their flights in bad weather will still get caught in it occasionally. The engineer must, therefore, face the job of producing airplanes that do not become hazardous in bad visibility. We cannot depend on skillful pilot techniques only to avoid this hazard.

Another consideration in making the airplane safer for all pilots is that strength must not be a prerequisite for proper handling of an airplane, either in the air or on the ground. The airplane must be capable of being safely and conveniently handled in all situations by, for instance, a woman weighing a mere 100 lb.

Since this discussion concerns simplifying the operation of an airplane rather than the simplification of the airplane, a part of the problem is to make an airplane that discourages the pilot from becoming careless. Methods of attack on this problem are:

1. Provide sufficient visibility forward and down that obstacles likely to produce a collision force themselves on the pilot's attention.

2. So relate the components of the airplane to the pilot that no accident is likely to be caused by his careless motions. This will include such things as separating motions in the controls so that in moving one control the pilot will not inadvertently move another (this seems to get back to coordination). It is also necessary to arrange the interior so that careless actions by passengers will not produce accidents. This requires that important controls be not accessible to passengers.

3. Reduce the number of controls so that the number of things the pilot has to do in flying will be reduced to the minimum, thus eliminating opportunities for carelessness.

An airplane will now be described in general terms which incorporates proposed solutions for most of the problems. This is in the general form of a specification, as follows:

A. Tricycle landing gear.

1. Steerable front wheel operated by aileron control.

2. Brakes on rear wheels operated by a single brake pedal.

3. Large rolling radius on front wheel (possibly same size as rear wheel).

4. Zero aerodynamic directional stability around rear wheels.

5. Sufficient shock absorption to permit landing without flare.

B. Stallproof.

1. Restricted elevator control.

2. Small center of gravity travel (necessary so that elevator control restriction will be sufficient for all center-of-gravity positions).

3. High longitudinal stability.

4. High damping in pitch.

5. Low moment of inertia in pitch.

C. Spinproof.

1. No independent control in yaw (two-control).

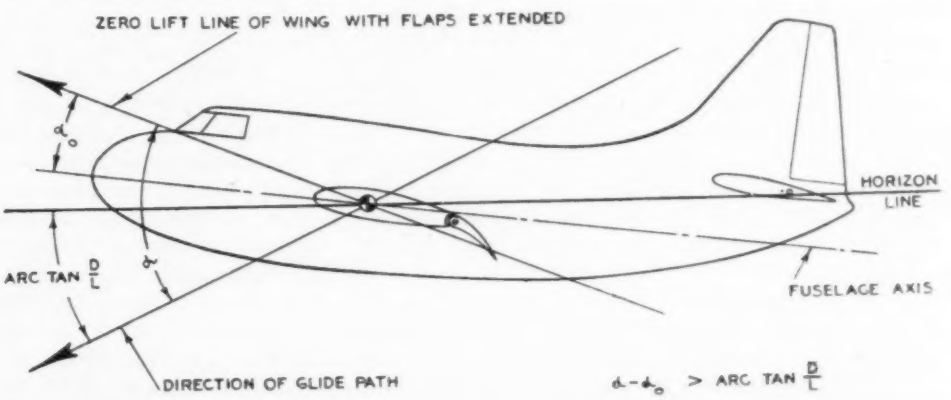
2. Low moments of inertia around all axes.

3. Principal axis of inertia in roll at a substantial negative angle to wing zero lift.

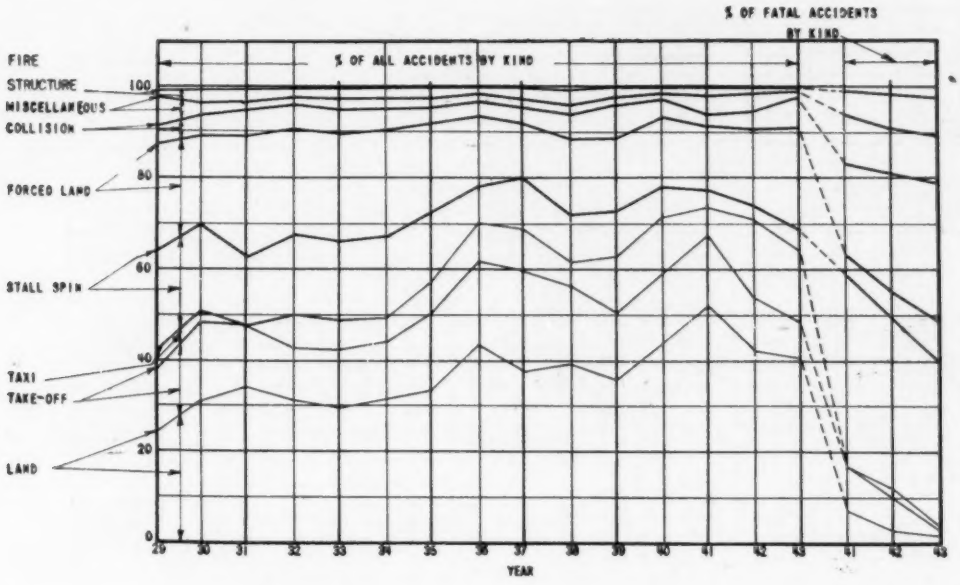
4. Heavy damping in yaw.

5. Not too great directional stability.

6. Completely satisfactory lateral control at the stall (stalls should be symmetrical). Aileron should have minimum change in section drag with aileron deflection.



▪ Fig. 11 – Attitude in glide desired for safety



■ Fig. 9 – Distribution of accidents by kind

conforms to the order previously stated. It is to be noted that as the utility of the airplane is increased and the private owner does more and more cross-country flying, the hazards from weather will increase.

The solution of the control problem requires a reduction in the complexity of control by eliminating unnecessary combinations of motions. The simplest conception is that of an airplane which always goes in the direction pointed, both vertically and laterally. This produces three degrees of freedom – turning, change in elevation, and change in speed, but one more motion than that of an automobile.

This conception is quite satisfactory in the air, but creates difficulties in landings and take-off. Even though in still air the airplane glides in the direction pointed, in a wind it does not glide in the direction pointed. The pilot, therefore, is not saved the problem of developing judgment of glide in winds of various velocities. Furthermore, in landing, due to cross wind, the airplane does not point in the direction traveled relative to the ground.

Assuming for the moment that the pilot can develop sufficiently good judgment of glide, he will collide with the ground nose downward if he does not develop depth perception. This is the most hazardous type of accident, because it produces deceleration forces parallel to the longitudinal axis of an airplane.

Neither the airplane structure nor the occupants are well suited for forces in this direction. The damage to people is high because they are thrown forward against whatever structure is ahead of them. This type of accident produces a high rate of personal injury.

The conception of an airplane that goes in the direction pointed will, therefore, be modified to one which goes in the direction pointed laterally, but is substantially level longitudinally at all times, or one in which, if exact longitudinal level cannot be achieved, deviations shall be in the direction of nose up, not nose down.

The pilot must still learn to judge glide, but at least he will not be fooled by the thought that he is gliding toward the point on the ground toward which the airplane is pointing.

This will then bring home to him that the throttle is still the altitude control, and he will come in with part throttle, and use throttle for control of his glide angle. No added glide control need be, nor should be, added. If it is, control is made more complex, not simpler, and the real problem is not to control the glide angle but to *know in time whether the angle of glide* is wrong.

A major advantage of keeping the airplane level, including during the glide, is that it can be flown into the ground without necessity for flare. The airplane structurally should be designed for this type of landing, and if so designed will eliminate the necessity for depth perception.

Naturally as pilots become more skilled, they will make a slight flare before landing to reduce landing loads on their airplanes. But by making flares unnecessary, a great deal will be accomplished to eliminate landing accidents both by reducing skill and judgment required in landing and by eliminating nose-down contacts with the ground.

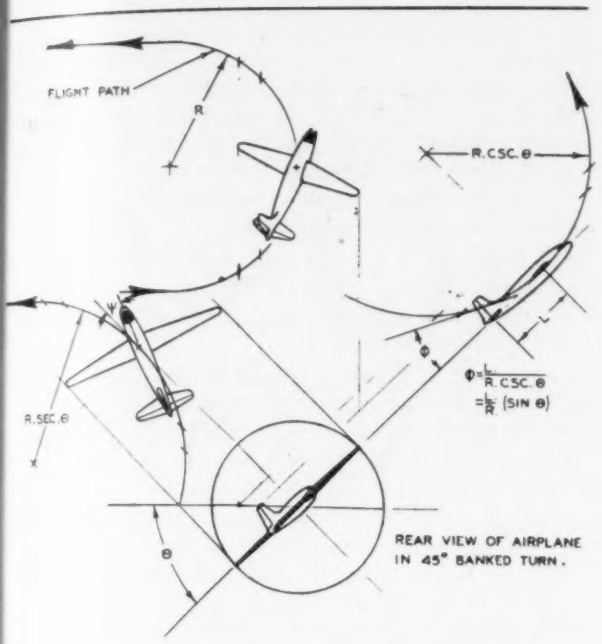
The problem of drift in cross-wind landings can be approached from either of the following two standpoints:

1. Require the pilot to detect it and give him means for correcting it just prior to landing.

2. Make an airplane which automatically corrects its heading on landing and which is capable of taking the loads generated by this action.

To solve the visibility problem, an airplane is required which can fly continuously at a reasonably low speed with complete control and with satisfactory engine cooling. With present techniques, 50 mph should satisfy this requirement, although 45 would be much better. An airplane with this low a flying speed can fly contact over known terrain in rather bad visibility. Furthermore, it will permit approaching landing areas at low altitude and low speed. With such an approach to the edge of the airport, the airplane can be landed by closing the throttle and permitting the airplane to settle on the airport in a horizontal position without change of air speed.

The visibility problem will certainly be increasingly solved by the use of radio and radar, but there will always be a need for contact flight in poor visibility. The visi-



▪ Fig. 10 – Constant altitude turn with inward slip

bility hazard cannot be controlled by regulations, because regulations of this type are not enforceable. As private flying grows, the policing problem alone will be impossible. People resent and disobey laws of the type which attempt to protect them from themselves.

Even those who do not begin their flights in bad weather will still get caught in it occasionally. The engineer must, therefore, face the job of producing airplanes that do not become hazardous in bad visibility. We cannot depend on skillful pilot techniques only to avoid this hazard.

Another consideration in making the airplane safer for all pilots is that strength must not be a prerequisite for proper handling of an airplane, either in the air or on the ground. The airplane must be capable of being safely and conveniently handled in all situations by, for instance, a woman weighing a mere 100 lb.

Since this discussion concerns simplifying the operation of an airplane rather than the simplification of the airplane, a part of the problem is to make an airplane that discourages the pilot from becoming careless. Methods of attack on this problem are:

1. Provide sufficient visibility forward and down that obstacles likely to produce a collision force themselves on the pilot's attention.

2. So relate the components of the airplane to the pilot that no accident is likely to be caused by his careless motions. This will include such things as separating motions in the controls so that in moving one control the pilot will not inadvertently move another (this seems to get back to coordination). It is also necessary to arrange the interior so that careless actions by passengers will not produce accidents. This requires that important controls be not accessible to passengers.

3. Reduce the number of controls so that the number of things the pilot has to do in flying will be reduced to the minimum, thus eliminating opportunities for carelessness.

An airplane will now be described in general terms which incorporates proposed solutions for most of the problems. This is in the general form of a specification, as follows:

A. Tricycle landing gear.

1. Steerable front wheel operated by aileron control.

2. Brakes on rear wheels operated by a single brake pedal.

3. Large rolling radius on front wheel (possibly same size as rear wheel).

4. Zero aerodynamic directional stability around rear wheels.

5. Sufficient shock absorption to permit landing without flare.

B. Stallproof.

1. Restricted elevator control.

2. Small center of gravity travel (necessary so that elevator control restriction will be sufficient for all center-of-gravity positions).

3. High longitudinal stability.

4. High damping in pitch.

5. Low moment of inertia in pitch.

C. Spinproof.

1. No independent control in yaw (two-control).

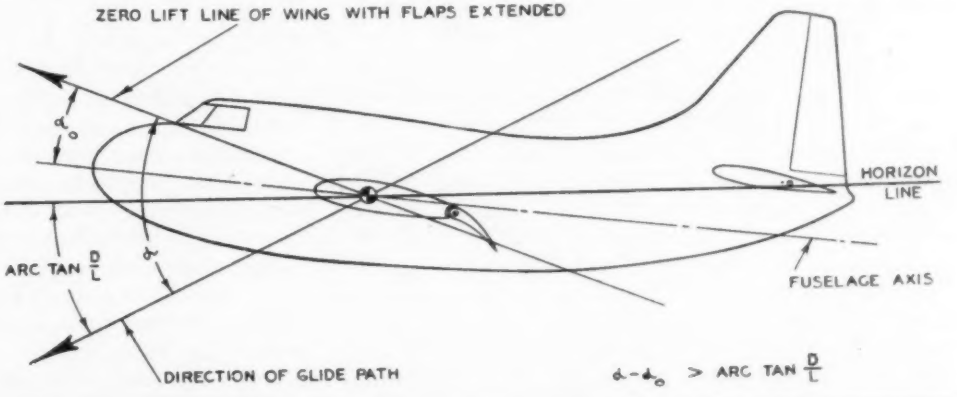
2. Low moments of inertia around all axes.

3. Principal axis of inertia in roll at a substantial negative angle to wing zero lift.

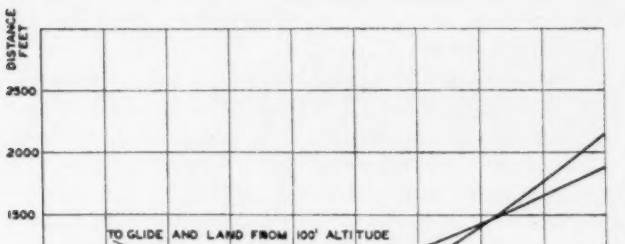
4. Heavy damping in yaw.

5. Not too great directional stability.

6. Completely satisfactory lateral control at the stall (stalls should be symmetrical). Aileron should have minimum change in section drag with aileron deflection.



▪ Fig. 11 – Attitude in glide desired for safety



■ Fig. 12 – Visibility requirements versus cruising speed for contact flying

7. Propeller located so that there are no unsymmetrical fin areas within the slipstream.

D. 50 mph minimum flying speed with complete control and engine cooling.

1. Same minimum speed power on as power off.
2. Climb available full throttle at minimum speed.

E. Automatic flap, down at low speeds, up at high speeds.

1. Airplane substantially level at all speeds.
2. Flap raised automatically when brakes are used.

F. Unobstructed vision forward and down.

1. Engine not in front (the forward engine location also is not compatible with low moments of inertia in pitch and yaw).

G. Design features, control arrangements, and eliminations to avoid carelessness.

1. Nonicing induction system (fuel injection).
2. Fuel system requiring no shut-off cock.
3. Control system arranged so that motion of one control does not produce inadvertently motion of another. This suggests foot accelerator for emergency application of power, and hand throttle for cruising (note that the feet are no longer busy operating the rudder).

The operation of this airplane will use the conditioned reflexes of an automobile driver to the maximum possible extent. On the ground it will control with exactly the same motions as an automobile: steer by a steering wheel, brake by a single pedal, and speed up and slow down by throttle action, preferably actuated by a foot accelerator.

The large rolling radius of the front wheel will reduce excessive nose wheel drag now encountered in mud and snow with small airplanes having very small nose wheels.

Zero aerodynamic directional stability around rear wheels will mean that in cross winds there will be no great yawing forces on the ground, and that the airplane will, therefore, taxi with ease in any direction relative to the wind.

In taking off the airplane will gain speed on the ground using the same technique as driving a car, and since the flap is automatically down, excessive up elevator will not be necessary to obtain sufficient angle of attack for take-off. This will permit keeping sufficient weight on the front wheel until take-off is reached, to give steering control on the ground.

The more experienced pilot, if he chooses, may press the brake pedal during the take-off run sufficiently to hold the automatic flap in the up position, but not sufficiently to apply brakes. This technique is of particular value in taking off in strong cross winds where a maximum steering control on the front wheel is desired.

The climb of this airplane will not differ from that of a conventional airplane, except that the nose will not have to be up appreciably more than the angle of climb, due to the action of the automatic flap.

Because the propeller is located so that there are no unsymmetrical fin areas in the slipstream, there will not be a variable yawing tendency due to power. The elimination of this variation in yawing with power is one of the most difficult problems in the two-control airplane.

In cruising, and perhaps in climbing, the hand throttle will be used rather than the foot accelerator. The presence of a foot accelerator, however, is of great use when temporary bursts of power are desired, as they may be obtained without disturbing the cruising setting of the throttle.

Automobile drivers all have a well-conditioned reflex for application of power in emergencies with a foot accelerator.

The use of a foot accelerator in such emergencies is in conformity with the campaign to eliminate chances of error. The pilot, in moving his foot to give more power, does not tend to move his upper body and arms, and thereby unconsciously move the aerodynamic controls. Further, there are more people who can, without looking, find a foot accelerator than there are who can, without looking, find a hand throttle.

The phenomena of going into a turn, maintaining a turn, and coming out of a turn need to be well understood before embarking on a discussion of simplifying the technique of control.

As has been pointed out, the curved flight path of a turn is produced by a horizontal component of lift produced by banking. The uses of the rudder in a turn are three. First, to *prevent* yawing of the airplane away from the turn, due to any adverse aileron yaw in entering the turn. Second, in the steady turn to *prevent* yaw of the airplane relative to the flight path. Third, in coming out of the turn to *prevent* yaw due to aileron action. The rudder does not produce the turn, but in all cases is used to *prevent* what has been considered, in a tradition of precision, *undesirable yaw*.

It is shown in the Appendix that the yawing moment of ailerons due to the change in induced drag is in a favorable direction. It is further shown in the Appendix that if the principal axis of inertia in roll makes a substantial negative angle with the zero lift line of the wing, favorable yawing is produced by roll.

The problem then of adverse yawing due to aileron is a problem of *change of section drag* of the airfoil with aileron deflection. This problem is not difficult. One simple solution is an aileron of such design that its deflection produces no convex discontinuity of contour at the aileron hinge.

In the steady turn, opposite aileron is frequently required to prevent overbanking. This is due to increased velocity of the outer wing tip increasing outer wing lift. Opposite rudder may or may not be required with this opposite aileron, depending on the yawing moments of the aileron.

The equilibrium of turning forces in the steady turn is

approximately as follows: First, due to damping in yaw, there is a yawing force tending to stop the turn. Second, there must be obtained from somewhere a yawing force into the turn to balance the damping in yaw. If the ailerons are the type having adverse yaw due to section drag, they themselves may, in those cases where opposite aileron is required in the steady turn, furnish part or all of the required yawing moment in a steady turn. If a further yawing force is required, rudder into the turn is used. This rudder into the turn may be required both in entering the turn and in the steady turn, whereas the aileron is required into the turn at the beginning of turn, but frequently opposite to the turn during the steady turn. This makes the problem of mechanical interconnection of the rudder and aileron rather difficult, there being no fixed mechanical ratio of motions between aileron and rudder that is correct for all conditions. Figs. 5, 6, and 7 previously shown illustrate this problem.

There is another available source of yawing moment into a turn, namely, inward slip plus directional stability. With inward slip, if the airplane is directionally stable, there will be produced a yawing moment in the direction of slip. Some airplanes tend to drop the nose in a turn and, therefore, require opposite rudder. This may be taken as an indication that the airplane has excessive directional stability, and that, therefore, the yawing moments accompanying any inward slip exceed the damping moments.

A roundabout conclusion may be made from these observations as follows: The amount of inward slip to produce a given yawing moment to overcome damping in yawing is in inverse proportion to the directional stability. Therefore, the less stable directionally the airplane, the more inward slip will be required in a turn.

Inward slip may be produced in a turn without loss of altitude by keeping the nose of the airplane up, as shown in Fig. 10. Not only is the required amount of inward slip inversely proportional to the directional stability, but it is also in direct proportion to the damping in yaw because the damping in yaw is the moment which must be opposed by the inward slip.

It may, therefore, be concluded that the way to keep the nose of an airplane up in a turn (remember, we are making an effort to design an airplane that will not strike the ground nose downward) is to have high damping in yaw, and low directional stability. With such a combination a mechanically interconnected rudder becomes unnecessary and the cost of the rudder and its control system is saved. By omitting the rudder and controls, a two-control airplane may be made more cheaply than a conventional airplane.

In making cross-wind landings with a two-control airplane, the problem of correction for drift immediately presents itself. Since the two-control airplane has no independent control in yawing, *any attempt* on the part of the pilot *to correct drift produces banking*. This results in landing on one wheel in a manner that aggravates the overturning tendencies due to the cross-wind landing. It is, therefore, necessary for the engineer to design the airplane in which no corrective action is required in a cross-wind landing, and in which the drift is automatically corrected by the airplane. It is also necessary that *the pilot* understand this and *not try to correct drift in cross-wind landings.*

It is my impression that the difficulties which have been encountered in cross-wind landings with those two-control airplanes now on the market have been encountered principally by experienced, conventional pilots, whose every instinct tells them to correct drift in a cross-wind landing, and who frequently get into trouble trying to do so. The inexperienced pilot, not seeing a difficulty, makes no effort to correct it, and the airplane, being capable of coping with the difficulty, straightens itself out without trouble.

Aside from the use of the tricycle gear, with its dynamic directional stability, there is a further important factor to be considered in the design of an airplane to correct drift automatically in cross-wind landings. This is aerodynamic directional stability on the ground.

Prior to a cross-wind landing, the airplane is flying, *relative to the air,* in the direction pointed, without yaw. When its wheels touch the ground, the drift is stopped and the airplane then has yaw relative to the air.

If the airplane is directionally stable about the rear wheels, this yawing moment will be up-wind. The correcting moments due to dynamic directional stability of the tricycle gear are down-wind, as this is the direction of the drift. Therefore, if the directional stability around the rear wheels is too great, the airplane will not correct for cross-wind drift but will head into the wind and ground loop.

It is for this reason that the specification for the airplane requires zero directional stability around the rear wheels.

The campaign to keep the nose up includes also keeping the nose up in all level flying and gliding speeds. This is accomplished by an automatic flap.

The answer in this case is a compromise, but it can be achieved with sufficient accuracy for the purpose. The flap action should shift the zero lift line of the wing in such a way that at a very low glide speed, which we may even call the landing speed but not the stalling speed, the required angle of attack of the fuselage is equal to or slightly greater than the glide path angle, as shown in Fig. 11.

At some slightly greater speed, called minimum cruising speed, at which it may be desired to cruise in conditions of bad visibility, the required angle of attack of the fuselage should be zero to slightly positive. This relationship turns out to be approximately that of an airplane with a fixed flap.

As the speed increases from the assumed minimum cruising speed, the fuselage angle of attack should remain sensibly constant. Since the wing angle of attack must be decreasing, the flap must be rising. This then is an automatic flap.

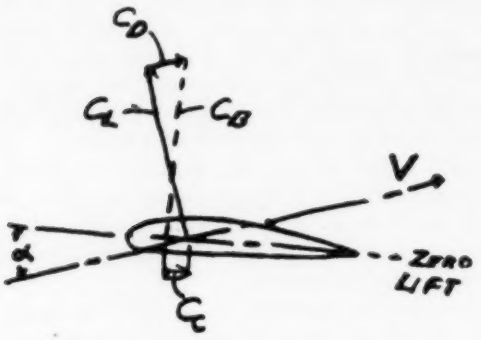
At high cruising speeds, used in good visibility, the change in angle of attack for changes in speed is very small, and it is satisfactory for the flap to be fixed in this condition.

We have thus written the specification for the automatic flap, namely, that it be all the way down between landing speed and minimum cruising speed, and shall be all the way up at a speed slightly less than cross-country cruising speed.

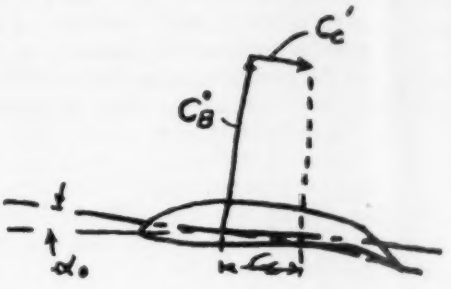
With the flap control eliminated, flight maneuvers will require motions quite similar to those required to drive a car, plus the added motion of the elevator.

It is possible that at some future date we may find it possible to fly cross-country on fixed elevator using high stability of the airplane to maintain trimming speed.

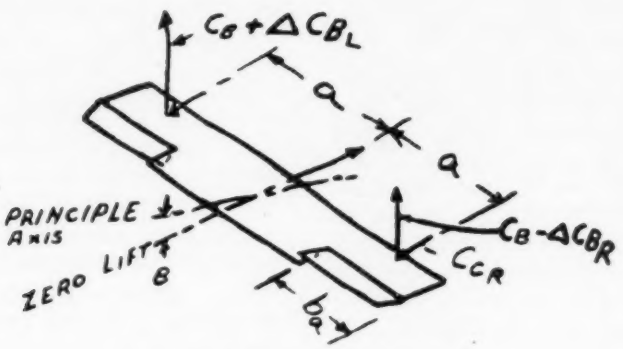
This is a subject well worth some research. There are two difficulties, both of which may turn out to be minor.



■ Fig. 13 – Forces on wing resolved to zero lift axes



■ Fig. 14 – Forces on wing resolved to axes at angle to zero lift



■ Fig. 15 – Rolling and yawing moments

One is that in an emergency elevator control should be instantaneously available. This is a mechanical problem easily solved.

The other is that with fixed elevator, the air speed will tend to increase in a turn. This is a distinct advantage with restricted up-elevator in that if the airplane is being flown at its absolute minimum speed with elevator all the way up against the stops, a turn will produce an increase in speed and entirely eliminate any hazard of stalling in the turn.

This is due to the component of curvature of the flight path in pitch which, in effect, is equivalent to reducing the up-elevator a slight amount. Expressed mathematically, this is equivalent to moving the zero lift line of the horizontal tail surfaces by an angle ϕ, as shown in Fig. 10:

$$\phi = L \sin \theta / R$$

where:

R = Radius of turn
θ = Angle of bank
L = Tail length

In cruising with fixed elevator, this may possibly be a handicap, as with fixed throttle altitude will be lost in a turn, for not only is more power required in a turn at constant air speed, but the increased air speed will also require more power. Possibly the solution for this problem will be to educate the pilot to apply increasing power in turns by using the foot accelerator so that when the turn is complete and he releases the accelerator, the airplane automatically returns to its trim condition.

The problem of approach to the airport requires the greatest amount of judgment because there is entailed not only the judgment of the glide of the airplane but also of the change in this glide path caused by the wind.

The accident rate from misjudged glide path angles and from misjudged altitude (depth perception) in landing approaches is very high. In the years 1941, 1942, and 1943, landing accidents averaged 45% of all accidents, and of the landing accidents 66% were due to poor technique, 6.3% due to lack of judgment, 10.2% due to carelessness.

It is proposed that landing glide and landing technique be simplified. The airlines long ago gave up the traditional aviation glide technique which required a high degree of skill and judgment, namely, squaring away at a distance from the airport and gliding down to the airport, engine idle, every landing being a rehearsal for a spot forced landing.

Airlines now approach with part power, with a continuous use of throttle for glide control, and there is no reason why the private owner pilot should not use the same technique. The nearer the approach speed is to the landing speed, the simpler will be the approach and landing technique, and the smaller the airport that can be successfully negotiated.

An airplane with satisfactory control down to the stall and also restricted elevator control to prevent the stall can safely be flown power on into the airport boundary at a speed of 5 to 10 miles above the minimum controllable flying speed. This is particularly true if a foot throttle is available for emergency bursts of power.

Such an approach permits the pilot to reach the airport

boundary at a minimum altitude and speed, at which point the throttle is closed. If the airplane is then maintained approximately level, the airplane will glide into the airport with no great judgment required as to glide path.

With the landing gear designed to take a "glide in" landing, it will not be necessary even for him to have depth perception or to flare. As the pilot's skill and judgment increase, he may improve on this technique by coming in a little higher, closing his throttle a little sooner so that the airport boundary is crossed in a steady glide, and thus use less of the airport in the glide.

It will be, however, no longer necessary for the pilot to *develop this skill and judgment before he can fly safely.*

We have covered the principal maneuvers of flight – taxiing, take-off, turning, and landing. There remains besides the problem of navigation, which will not be discussed, the problem of safe flying in bad visibility.

In the belief that instrumental techniques for blind flying are not within the skill nor the pocketbook of the average private owner, it is proposed that the engineer's job is to increase continuously the safety of flying in bad visibility. This can be accomplished by reducing the minimum controllable flying speed.

The required visibility range for safe flying is a function of two things – the pilot's reaction time x the airplane speed, and the minimum available radius of turn, which varies with the square of the flying speed. It is apparent then that the ability to slow down will greatly increase safety as visibility decreases.

I am not suggesting that we encourage people to take off in dangerous weather, but I am suggesting that we recognize the fact that people will take off in dangerous weather, and will get caught in dangerous weather, and it is our job to help them.

If we assume that the pilot's reaction time is 1 sec and that the minimum radius of turn is that accompanying a 45-deg bank, and that the distance to generate a 90-deg turn of minimum radius is 3 times the turn radius, we find that the visibility required to avoid collision, expressed as a function of speed, equals $1.466\ V + 0.2V^2$.

If it be assumed that in bad weather, contact flight search for a landing area over known terrain can be made at 100-ft altitude, that the airplane has a sinking speed of 10 fps in a glide at minimum cruising speed, and that the braking deceleration is 0.8 g, the distance required to glide in at minimum cruising speed and stop is $14.66\ V + 0.0417\ V^2$.

These two equations are plotted in Fig. 12. Note that below 83 mph, it takes a longer distance to land than to maneuver, and at this speed, it takes about 1510 to do either. At 50 mph, 570 ft are required for maneuvering and 840 ft for landing. Thus, below 83 mph the pilot is safe looking for a landing area because he can find one without running out of visibility required for maneuvering.

In summing up, the problem resolves itself into these factors:

1. Eliminate need for coordination.
2. Eliminate need for depth perception.
3. Eliminate nose-down contact with ground.
4. Reduce minimum cruising speed to decrease need for visibility.
5. Reduce landing speed to reduce kinetic energy in landing.
6. Reduce temptation to carelessness.

Or, in basic philosophy, let us cease trying to mold people to fit the airplane, but let us fit the airplane to people as they are.

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2. CAR 04, Nov. 1, 1943.
3. CAR 03, November, 1945.
4. CAA Statistical Handbook, Oct. 15, 1944.
5. NACA Technical Report No. 457 (1933), "Maneuverability Investigation of O3U-1 Observation Airplane," by F. L. Thompson and H. W. Kirschbaum.
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Appendix

Yaw Due to Ailerons

Part I

Forces on a wing resolved to zero lift axes (see Fig. 13):

$$C_B = C_L \cos\alpha + C_D \sin\alpha$$

$$C_e = C_D \cos\alpha - C_L \sin\alpha$$

$$\alpha = C_L\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)$$

Let $\sin\alpha = \alpha \cos\alpha = 1 - \frac{\alpha^2}{2}$

$$C_B = C_L\left(1 - \frac{\alpha^2}{2}\right) + \left(C_{D_0} + C_{D_i}\right)\alpha$$

$$C_B = C_L\left[\frac{1-C_L^2}{2}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)^2\right] + \left(C_{D_0} + \frac{C_L^2}{\pi R}\right) C_L\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)$$

$$C_B = C_L\left[1 - C_L^2\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)\left(\frac{1}{2a_0} - \frac{1}{2\pi R}\right) + C_{D_0}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)\right]$$

$C_B = C_L$ (approximately) (1)

$$C_e = \left(C_{D_0} + C_{D_i}\right)\left(1 - \frac{\alpha^2}{2}\right) - C_L\alpha$$

$$C_e = \left(C_{D_0} + \frac{C_L^2}{\pi R}\right)\left[1 - \frac{C_L^2}{2}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)^2\right] - C_L^2\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)$$

$$C_e = C_{D_0} - \frac{C_L^2}{a_0} - \frac{C_L^2}{2}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)^2\left(C_{D_0} + \frac{C_L^2}{\pi R}\right)$$

$$C_e = C_{D_0} - \frac{C_L^2}{a_0} \text{ (approximately)} = C_{D_0} - \frac{C_B^2}{a_0} \quad (2)$$

From (1) and (2) it may be concluded that C_B and C_e are inherent to the airfoil, and that aspect ratio determines only the angle of attack required to obtain a particular C_B.

Part II

Forces on a wing resolved to axes at angle α_0 to zero lift (see Fig. 14):

In considering the case of a wing with flap or aileron deflected, it is convenient to refer the forces to the zero lift line with flap neutral.

Let C_B' and C_c' refer to zero lift line, flaps deflected, and C_B and C_c refer to original zero lift line. Let $C_B' = C_B$.

$$C_c = C_c' + C_B'\alpha_0$$

$$C_c = C_{D_0} - \frac{C_B^2}{a_0} + C_B\,\alpha_0 \qquad (3)$$

Part III

Rolling and yawing moments (see Fig. 15):

Assume ailerons deflected to shift zero lift lines by angles α_L and α_R:

$$\alpha_L = \Delta C_{B_L}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)$$

$$\alpha_R = \Delta C_{B_R}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)$$

Let increments of section drag due to aileron deflection be $\Delta C_{D_{0L}}$ and $-\Delta C_{D_{0R}}$.

$$C_{c_L} = C_{D_0} + \Delta C_{D_{0L}} - \left(\frac{C_B + \Delta C_{B_L}}{a_0}\right)^2 + \left(C_B + \Delta C_{B_L}\right)\Delta C_{B_L}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)$$

$$C_{c_R} = C_{D_0} - \Delta C_{D_{0R}} - \left(\frac{C_B - \Delta C_{B_R}}{a_0}\right)^2 - \left(C_B - \Delta C_{B_R}\right)\Delta C_{B_R}\left(\frac{1}{a_0} - \frac{1}{\pi R}\right)$$

Rolling moment $= C_l\,Sbq = cb_a\,aq\left(\Delta C_{B_L} + \Delta C_{B_R}\right)$

$$C_l = \frac{cb_a\,a}{Sb}\left(\Delta C_{B_L} + \Delta C_{B_R}\right) \qquad (4)$$

Yawing moment $= C_n sbq = cb_a\,aq\,(-C_{c_L} + C_{c_R})$

$$C_n = \frac{cb_a a}{sb}\left[-C_{D_0} - \Delta C_{D_{0L}} + \left(\frac{C_B + \Delta C_{B_L}}{a_0}\right)^2 - \left(C_B + \Delta C_{B_L}\right)\Delta C_{B_L}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right) + C_{D_0} - \Delta C_{D_{0R}} - \left(\frac{C_B - \Delta C_{B_R}}{a_0}\right)^2 - \left(C_B - \Delta C_{B_R}\right)\Delta C_{B_R}\left(\frac{1}{a_0} + \frac{1}{\pi R}\right)\right]$$

Dropping $\Delta C_B{}^2$ terms and combining:

$$C_n = \frac{cb_a a}{sb}\left[-\Delta C_{D_{0L}} - \Delta C_{D_{0R}} + C_B\left(\Delta C_{B_L} + \Delta C_{B_R}\right)\left(\frac{1}{a_0} - \frac{1}{\pi R}\right)\right] \qquad (5)$$

Note that $\frac{1}{a_0} > \frac{1}{2\pi}$ and $\frac{1}{a_0} - \frac{1}{\pi R} > 0$ for $R > 2$.

Therefore, referenced to normal wing zero lift axes the yaw due to roll is favorable, the only unfavorable yaw being due to change of section drag with aileron deflection.

Part IV

Effect of inclination of principal axes of inertia:

Let principal axis of inertia in roll be at angle θ to zero lift line, as shown in Fig. 15.

Let C_l', C_n', ϕ', and ψ' refer to principal axis of inertia.

$$C_l' = C_l\cos\theta + C_n\sin\theta$$

$$C_n' = C_n\cos\theta - C_l\sin\theta$$

$$\frac{d^2\phi'}{dt^2} = \frac{Sbq}{Ix}\left(C_l\cos\theta + C_n\sin\theta\right)$$

$$\frac{d^2\psi'}{dt^2} = \frac{Sbq}{Iz}\left(C_n\cos\theta - C_l\sin\theta\right)$$

These motions combine to give a yaw referenced to zero lift axes.

$$\frac{d^2\psi}{dt^2} = \frac{d^2\psi'}{dt^2}\cos\theta + \frac{d^2\psi'}{dt^2}\sin$$

$$= \frac{Sbq}{Iz}\left(C_n\cos^2\theta - C_l\sin\theta\cos\theta\right) + \frac{Sbq}{Ix}\left(C_l\sin\theta\cos\theta + C_n\sin^2\theta\right)$$

$$= \frac{Sbq}{Iz}\left[C_n\left(\cos^2\theta + \frac{Iz}{Ix}\sin^2\theta\right) + C_l\sin\theta\cos\left(\frac{Iz}{Ix} - 1\right)\right]$$

Substituting $C_l + C_n$ from (4) and (5):

$$\frac{d^2\psi}{dt^2} = \frac{cb_a q}{Iz}\left\{\left(\Delta C_{B_L} + \Delta C_{B_R}\right)\left[C_B\left(\frac{1}{a_0} - \frac{1}{\pi R}\right)\left(\cos^2\theta + \frac{Iz}{Ix}\sin^2\theta\right) + \left(\frac{Iz}{Ix} - 1\right)\sin\theta\cos\theta\right] - \left(\Delta C_{D_{0L}} + \Delta C_{D_{0R}}\right)\left(\cos^2\theta + \frac{Iz}{Ix}\sin^2\theta\right)\right\} \qquad (6)$$

Note that the favorable yaw due to roll is augmented by θ, together with the fact that usually $Iz > Ix$, not only by the increase in $\cos^2\theta + \frac{Iz}{Ix}\sin^2\theta$, but by the added term $\left(\frac{Iz}{Ix} - 1\right)\sin\theta\cos\theta$.

It is also evident that if ΔC_{D_0} is small enough, the whole yawing moment can be favorable, without requiring that ΔC_{D_0} be negative.

IMPROVING ENGINE PARTS

continued from page 475

the magnitude of the stress calculated from the anticipated imposed loads in relation to any one property or characteristic of the material used. For economical reasons, designs should be proportioned to take advantage of any prestress which can be beneficial. Recognition must be given to the properties of the material in the actual finished piece and these properties weighed against the service requirements.

Direct measurement of strain requires considerable research and study before its advantages can be evaluated. It has already proved its worth in strengthening many machine parts. It points the way to the economical distribution of material in the part, thus improving the design and contributing to progress. It seems certain that the field of usefulness of analysis by direct measurement of strain will be greatly enlarged in the near future.

BMW-003 TURBO-JET ENGINE

Compared with the JUMO 004

by W. G. LUNDQUIST and R. W. COLE
Wright Aeronautical Corp.

PERFORMANCE characteristics of the German BMW-003 and Jumo 004 engines as a whole, as well as of their principal parts, are presented here.

This paper, the authors point out, is not the result of experimental determinations in this country, but is rather a résumé of data obtained from German sources.

The discussion is divided as follows:

1. Performance of complete machine.
2. Compressor performance.
3. Combustion-chamber performance.
4. Turbine performance.
5. Miscellaneous operating characteristics.

IN order not to duplicate previously published data, this paper will omit practically all description of mechanical features of the machines under consideration, and will instead confine itself to the presentation of performance characteristics of the overall machines and the separate performance characteristics of their principal components. Furthermore, it should be pointed out at the beginning that the material presented herewith is not the result of experimental determination in this country, but is rather a résumé of data obtained from German sources. The data included in this paper were collected from the voluminous files of technical information obtained in Europe by Army and Navy technicians, and the authors wish to acknowledge the courtesy of the military forces in making this information available.

The very volume of the technical files at our disposal for this purpose has in itself posed a problem. As might be expected in an event such as that involved by the collapse of Germany, a great deal of conflicting information was accumulated as a natural result of having numerous investigators separately delving into the records of such a complex activity as that represented by the German aircraft powerplant field. A conception of the problem in this respect can be gained if we but imagine the task a strange investigating committee would have in ascertaining the latest status of our own development project in this country. In fact, we are hard pressed at times to explain to ourselves just where we are in the case of some of our more advanced projects, because of the kaleidoscopic changes which are characteristic of an industry which is progressing as rapidly as is the aircraft powerplant field. The authors make no pretense of having made a complete digest of data available, but they believe the following presents a fairly accurate statement of the actual performance obtained from the machines under consideration.

This paper is, therefore, not an analysis of performance but rather a compilation of principal performance characteristics intended for reference purposes. For convenience, the discussion will be subdivided as follows:

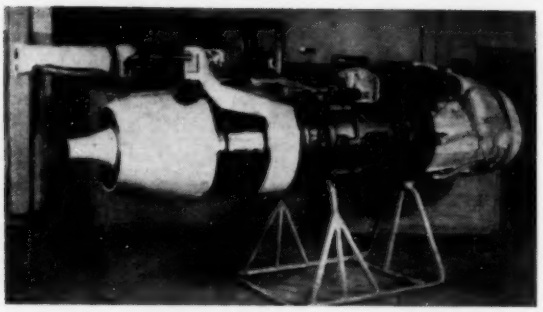
I. Performance of the complete machine.
II. Compressor performance.
III. Combustion-chamber performance.
IV. Turbine performance.
V. Miscellaneous operating characteristics.

Performance of Complete Machine

The machine with which we are principally concerned is the BMW-109-003 turbo-jet engine. Fig. 1 is a photograph of the rocket equipped model of this engine. To refresh our memories, this machine consists of a seven-stage axial-flow compressor, an annular combustion chamber, a single-stage turbine, and a variable orifice jet. The rocket attachment was not part of the basic machine but was an added component, as will be described later. The compression ratio is approximately 3.1 to 1, and the operating temperature is approximately 1380 F. This machine was just going

[This paper was presented at the SAE National Aeronautic Meeting, New York, April 3, 1946.]

THE AUTHORS: W. G. LUNDQUIST (M '29), chief engineer of Wright Aeronautical Corp., is a graduate of the University of Minnesota from which he received his B.S. in M.E. in 1928. With Wright since 1929, he has during this time been engaged in all phases of engine development. Much of the material presented in the paper on the BMW-003 jet engine was collected in late 1945 when he was in Germany as a member of the U. S. Naval Technical Mission in Europe. R. W. COLE (J '43) has been with Wright Aeronautical Corp. since 1941, during which time he has been engaged in turbosupercharger and gas turbine development, specializing in performance analysis. He taught for two years at New York University after his graduation from Rensselaer Polytechnic Institute in 1939.



■ Fig. 1 – BMW-109-003R turbo-jet rocket engine

into large-scale production during the latter stages of the war. For reference purposes, we have included in this paper some comparative performance characteristics of the Junkers 109-004 machine which had earlier reached production status.

At least nine different models of the BMW-003 turbo-jet engine are known to have been constructed, or at least designed, during the development process. Of these the 003A-2 was the one actually in production at the end of the war. Table 1 lists these known models with the available information on their performance characteristics.

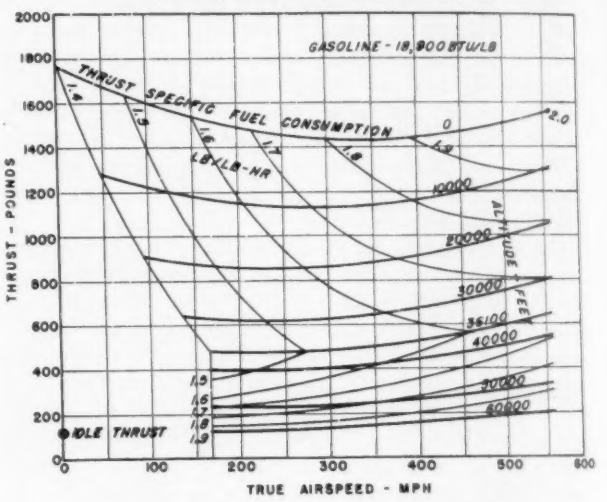
The 003A-0 was an early semiproduction model made in 1943 and used for initial testing and development. The 003A-1 was the first production model and most of the performance data were established by this machine. It was superseded by the A-2 final production model, which differed only in structural details contributing to durability and safety. Only one unit of the 003B-1 was made and this was abandoned in favor of the lighter and more efficient 003C model, which was scheduled to replace the 003A-2 in production. The C type employed an advanced compressor designed by the Brown-Boveri Co. of Mannheim. The 003D had not progressed beyond the layout and calculation stage, but was to be an entirely new design with added compressor and turbine stages and much better performance. The 003E-1 and E-2 differed from the production type only in the installation changes made for use in the He-162 airplane. The 003R model was a production type of turbo-jet with a nonexpendable liquid fuel rocket unit attached. This rocket provided an additional thrust of 2700 lb for 3 min maximum duration at approximately 20 lb per hr per lb of thrust specific fuel consumption. Although it could be used in take-off this was not primarily a jato rocket, but was intended to provide exceptional combat climb performance for interceptor fighters.

The Junkers Jumo 109-004B was the first German turbo-jet in active production. Since Junkers engines were supplying the demand quite well, production of the BMW-003A was delayed to permit further development. As a result the BMW machine was more durable, although surprisingly little performance improvement is apparent. Table 2 compares the performance and significant design dimensions of the two types, showing how closely they resemble one another. It is evident from this table that the Jumo 004 was a slightly larger machine, both physically and in the amount of thrust developed. However, the specific fuel consumptions were almost identical, as were the specific weight, the cycle temperature, and general component efficiencies. The only striking difference was the type of combustion chamber, since the 004 had six individual chambers, while the 003 used a single annular design. On the basis of the performance and physical dimensions there is little choice between these machines, but the development of the 003 was directed toward ease of manufacturing, operating safety, durability, and high-altitude burner performance, which factors do not appear in a tabulation of this type. In regard to these latter characteristics the 003 was reportedly superior and potentially it is believed it had a more promising future.

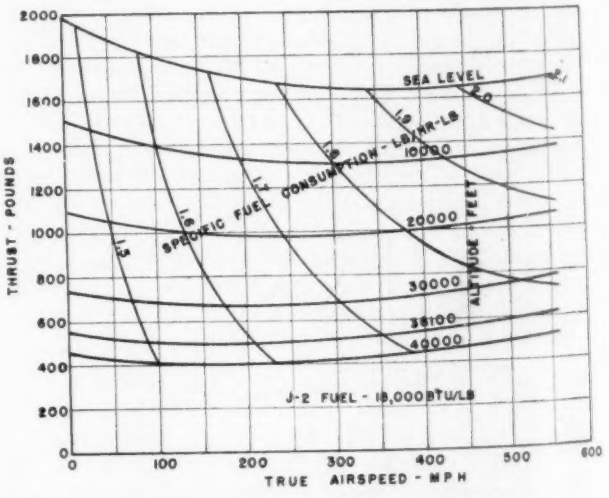
Fig. 2 shows the specification performance of the 003. These curves are drawn for maximum rpm and burner

9500 RPM COMBUSTION TEMPERATURE 1382°F

ALTITUDE FEET	FLIGHT SPEED MPH		THRUST LB	FUEL CONSUMPTION LB/LB-HR	FUEL CONSUMPTION LB/HR
0	0	STATIC THRUST	1764	1.4	2469
0	559	FLIGHT THRUST	1554	2.0	3108



■ Fig. 2 – Performance of BMW-003A engine



■ Fig. 3 – Performance of Jumo 004B engine – 8700 rpm – international normal atmosphere

Table 1 – Design Data of BMW-003 Series Turbo-Jets

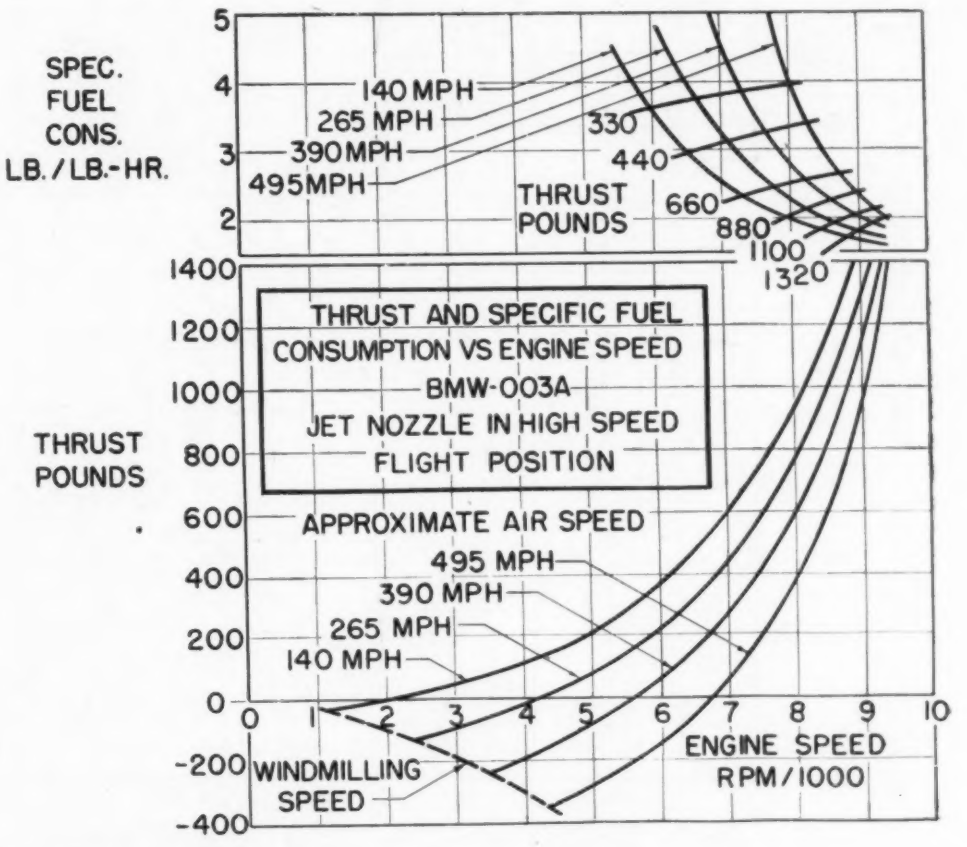
Model	Static Thrust, lb	Specific Fuel Consumption, lb per lb-hr	Engine Speed, rpm	Weight, lb	Airflow, lb per sec	Pressure Ratio	Compressor Stages	Compressor Tip Speed, fps	Turbine Stages	Turbine Tip Speed, fps
003A-0	1000[a]	2.2[a]	....	1650	....	....	....	....	....	....
003A-1	1764	1.47	9500	1345	....	....	....	....	....	....
003A-2	1764	1.47	9500	1345	41.9	3.09	7	8.98	1	1024
003B-1	1980	1.40	9500	1650	....	....	7	....	1	....
003C	1984	1.30	9800	1345[a]	44.1	3.42	7	9.35	1	1056
003D	2646	1.10	10000	1366[a]	55.1	4.91	8	9.61	2	1089
003E-1	....	....	....	....	....	....	....	....	....	....
003E-2	....	....	....	....	....	....	....	....	....	....
003R	1764	1.47	9500	1576	....	....	....	....	....	....

[a] Information doubtful.
Note: Operating temperature for all models – 1382 F.

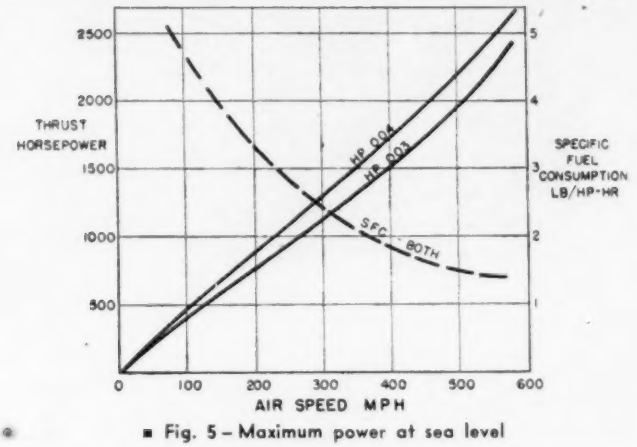
temperature on the assumption of an appropriately variable jet orifice and as such are quite conventional and self-explanatory. Note in this figure that a declining combustion efficiency was presumed above 36,100-ft altitude, causing the specific fuel consumption to become rapidly worse with increasing altitude. Had the same constant efficiency been used throughout, the lines of fuel specific in the isothermal region would have been vertical like the left-hand boundary line at 1.4 lb per lb of thrust. The low fuel specifics compared to Tables 1 and 2 are accounted for by the fact that the curve is based on the use of gasoline rather than J-2 fuel. It should be pointed out here that the Germans were forced by shortages of petroleum to use progressively poorer fuels in their jet engines. The first fuel, a good grade of gasoline, had a lower heating value of 18,900 Btu per lb, but later, the J-2 fuel, a light diesel oil type with a heating value of 18,000 Btu per lb, was substituted. Toward the end of the war tests were in progress to develop burners using crude petroleum but this had not reached production status. Fig. 3 shows the performance of the Jumo 004 turbo-jet, with fuel consumption based on the use of J-2 fuel.

Referring to the preceding curves, while the performance indicated required a variable jet orifice, the production 003 had only four manually selected positions, these positions being starting and idling, climb, high speed at low altitude, and high speed at high altitude. Because this arrangement had caused considerable difficulty in combat, an automatic control had been designed and was being tested when production stopped. The 004 unit had automatic control based on control lever selection of jet positions.

The variation of fuel consumption and thrust with rpm at several air speeds for the 003 is shown in Fig. 4. For the benefit of those who prefer to think of horsepower



▪ Fig. 4 – Variation of fuel consumption and thrust with rpm for BMW-003A – sea level test data

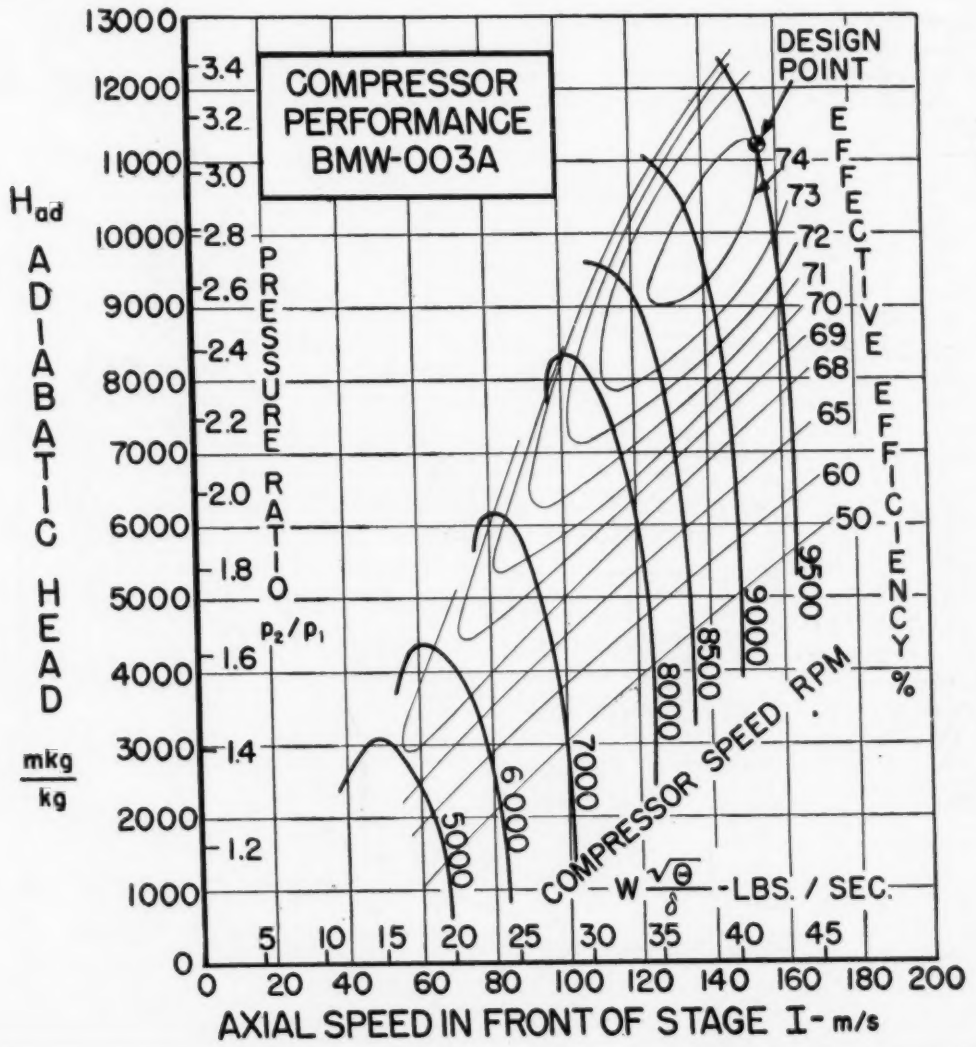


■ Fig. 5 – Maximum power at sea level

rather than thrust, Fig. 5 shows the thrust horsepower and power specific fuel consumption of both machines at sea level.

■ Compressor Performance

The compressor of the 003 is a seven-stage axial-flow type employing NACA airfoil sections. The pressure ratio per stage is approximately 1.17:1, giving an overall ratio of 3.09:1 at the sea level static design point. Most of the German technical papers on the subject have stressed the difficulty of matching compressor stages, and numerous tests of sections consisting of two or three stages were made to obtain the best matching of the components. The test of the complete unit with interstage measurements, however, shows that in spite of the extensive development, correct matching of the parts had not been achieved, and hence the overall efficiency curve is as shown in Fig. 6. This figure is a direct copy of a BMW test curve with the German method of presentation. The equivalent NACA scales are added for convenience in comparisons. The quantity "adiabatic head" is defined as the work required to compress isentropically one pound of air through the pressure ratio actually obtained. Thus, it represents the isentropic enthalpy change between the initial and final pressures. The ratio of this adiabatic work and the actual work as measured by shaft power is the effective efficiency shown on the curve. At standard sea level initial temperature the adiabatic head, adiabatic temperature rise $(Y - T_1)$, ideal work of compression, and pressure ratio are mutually convertible, while the ordinate, axial speed



■ Fig. 6 – Overall efficiency curve for compressor of BMW-003A engine

before Stage 1, is simply another volume flow term similar to $Q/\sqrt{\theta}$ or $W\sqrt{\theta}/\delta$. It should be emphasized that the efficiencies shown are based on shaft power measurements and therefore include mechanical losses. Fig. 7 shows the relative magnitude of efficiency calculated on this basis as compared to the conventional temperature rise ratio or thermal efficiency.

The German engineers interrogated quite freely admitted that they considered the 003A compressor performance rather poor, and this is confirmed by the fact that there were two other compressor designs under investigation for this machine. These were the BBC Mannheim design for the 003C and the BMW advanced design for the 003D, as indicated in Table 1. Tests had demonstrated that for the 003A the major losses occurred in the stators, which were approximately 30% reaction, and in an effort to improve the stator performance the later designs employed about 50% reaction. It is of interest to note that the Jumo 004 machine had almost pure impulse stators made of curved sheet metal, and as a result required one more compressor stage to obtain the same pressure ratio. However, the thermal efficiency of the 004 compressor was slightly better than that of the 003.

Combustion Chamber

The combustion chamber of the 003 engine is interesting because it is of the annular-chamber type. The Junkers 004 engine used individual burner cylinders – a form of combustion chamber which has gained many adherents both in England and in this country. The writers believe that the annular-chamber type will ultimately prove to be best from an efficiency standpoint. The BMW engineers had apparently arrived at the same conclusion, and it is interesting to follow the successive steps in their burner development.

In their preliminary combustion-chamber studies, the BMW engineers decided that the best way to provide stable combustion without blowout was to produce local eddy regions by installing baffles in the air stream. The three principal forms of these baffle plates are shown in Fig. 8.

Fig. 8a shows the first form tried. This consisted of 16 individual ceramic baffle plates mounted in the annular chamber as shown. The fuel nozzles were located upstream and the fuel sprays were allowed to impinge on the baffle plates. The total airflow passed around these baffles. This arrangement proved to be generally unsatisfactory because the airflow around the baffles was nonuniform around the annular chamber and the resulting temperature distribution downstream of the plates was bad. The blowout characteristics were also unsatisfactory.

The second form of baffle plate burners is shown by Fig. 8b. In this case each baffle plate was supplied with air through upstream projecting conical tubes. This resulted in considerable improvement in blowout characteristics, and made possible full-load airflow operation. Combustion efficiency and temperature distribution were still unsatisfactory. Both of the above forms also gave trouble because of breakage of the ceramic baffles.

In an attempt to correct these troubles, the ceramic baffle plates were replaced by perforated steel plates, with the arrangement otherwise the same as shown by Fig. 8b. The perforated steel plates further improved the blowout characteristics, and almost completely eliminated the baffle plate breakage, making possible almost unlimited baffle plate life. Combustion efficiency and temperature distribution, however, were still not satisfactory. To improve these latter characteristics, additional round baffle plates were installed farther downstream but no improvement resulted.

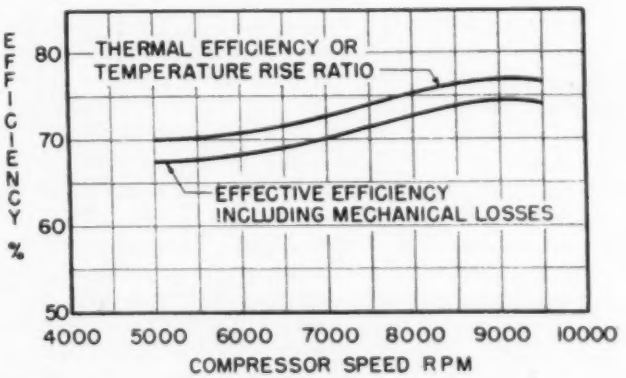
Table 2 – Comparison of 003 and 004 Production Turbo-Jets

	BMW-109-003A-2	Jumo 109-004B-4
Static Thrust, lb	1764	1984
Specific Fuel Consumption, lb per lb-hr	1.47	1.48
Turbine Speed, rpm	9500	8700
Weight, lb	1345	1543
Specific Weight, lb per lb thrust	0.76	0.78
Length, in.	143	163
Diameter, in.	27.2	28.7
Airflow, lb per sec	41.9	46.3
Compressor Tip Diameter, in.	21.65	21.46 (Stage 1)
Compressor Diameter Ratio	0.725	0.645
Number of Compressor Stages	7	8
Turbine Tip Diameter, in.	24.70	27.48
Turbine Diameter Ratio	0.71	0.68
Burner Temperature, F	1382	1390
Number of Burners	16	6
Combustion Chambers	1 annular	6 individual

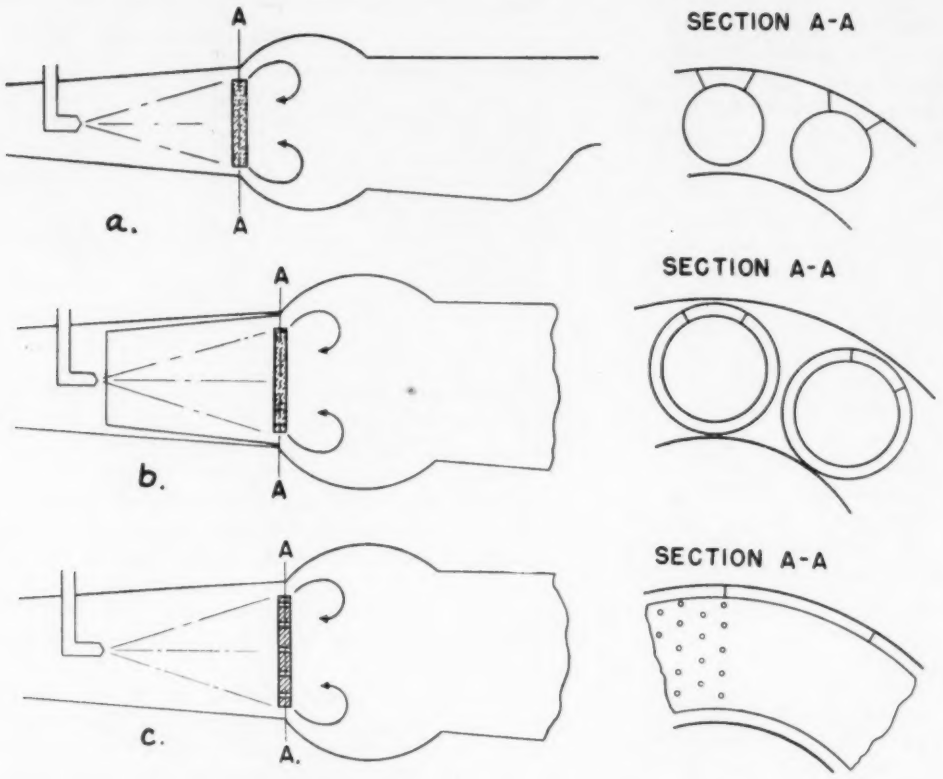
The final baffle plate arrangement tried is shown by Fig. 8c. This consisted of a completely annular perforated steel baffle. The theory behind this attempt was that such an arrangement would produce a complete annular ring of flame. The results were disappointing. The continuous flame ring did not materialize, and the pressure drop was raised considerably.

At this stage of the development, the BMW engineers decided that there was something basically wrong with the baffle-plate type of burner. Although blowout characteristics were passable, combustion efficiencies were only in the order of 60-70%. The maximum to mean temperature ratios were approximately 2.0:1. The flames were too long. These poor characteristics appeared to be caused by having the fuel impinge on the baffle plates and hence the remedy lay in injecting the fuel into the eddy regions downstream of the baffles. This was tried experimentally by installing the fuel nozzles downstream and spraying the fuel upstream into these eddy regions. The results apparently confirmed the theory. Combustion efficiencies were reportedly 20% better and the maximum to mean temperature ratio dropped to 1.5:1. This arrangement was considered the best of the baffle-plate type.

The baffle-plate type of burners did not go into production, however, in spite of the improvement noted above. While the experiments just described were going on, another means of producing the combustion-chamber eddy



■ Fig. 7 – Compressor peak efficiencies for BMW-003A engine



■ **Fig. 8 – Forms of baffle plates for combustion chamber**
(a) Combustion chamber with ceramic baffle plates
(b) Combustion chamber with air supply tubes
(c) Combustion chamber with eddy ring

zones was also being studied. This consisted of the conical eddy-producing burner shown in Fig. 9. With this type of burner, the fuel is sprayed directly into the toroidal eddy produced by the air flowing over the downstream edge of the burner cone. At the same time that BMW engineers adopted this type of burner, they also made another basic change in their attack on the burner problem by changing from a total to a partial airflow past these burner elements. To accomplish this, they separated the total airflow into primary and secondary streams, and passed only the primary air through the burner cones. The secondary air was diverted and introduced downstream, as shown in Fig. 10. The relative proportions of primary and secondary air were experimentally determined and a final distribution of 60-70% primary and 30-40% secondary air was selected as optimum. All this finally led to the production combustion chamber which is shown in Fig. 10.

Referring to Fig. 10, it is interesting to note the gas velocities and Mach numbers through the combustion chamber. The advantage of the annular chamber can be noted in the low Mach numbers prevailing. The low gas velocities make the momentum pressure loss negligible. A small amount of air is admitted around the fuel nozzle itself. This reduces the temperature peak at the center of the flame core and also cools the nozzle – preventing internal carbonization of the nozzle. The purpose of the secondary air nozzles is to introduce this air uniformly into the hot central core of the primary air stream. The final production combustion chamber reportedly had a combustion efficiency of 92-95%, with a total head loss of 0.15 atmospheres (3 atmospheres inlet). The maximum to mean temperature ratio was in the order of 1.2:1.

Fig. 11 shows the temperature distribution at the nozzle

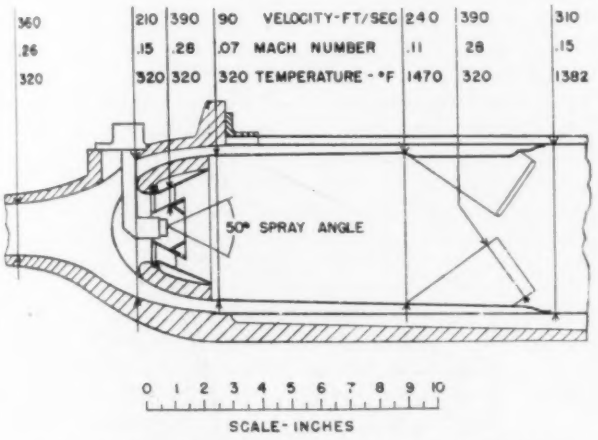
■ **Fig. 9 – Burner cone for BMW-003A**

end of the combustion chamber. This temperature distribution spread obviously is quite good.

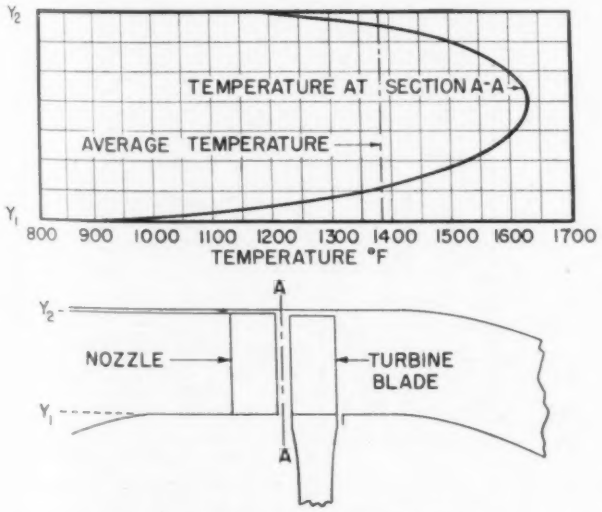
Fig. 12 shows the upstream end of the combustion-chamber casting, and shows the 16 holes for the primary air inflow to the burner cones, with the cones themselves visible in the background.

Fig. 13 shows the 80 secondary air nozzles looking upstream toward the burner cones. These nozzles extended into the hot flame core and operated at approximately 800 C at the tips. For this reason they were made of a special heat-resisting steel as contrasted with the rest of the combustion chamber, which was made principally of low carbon steel.

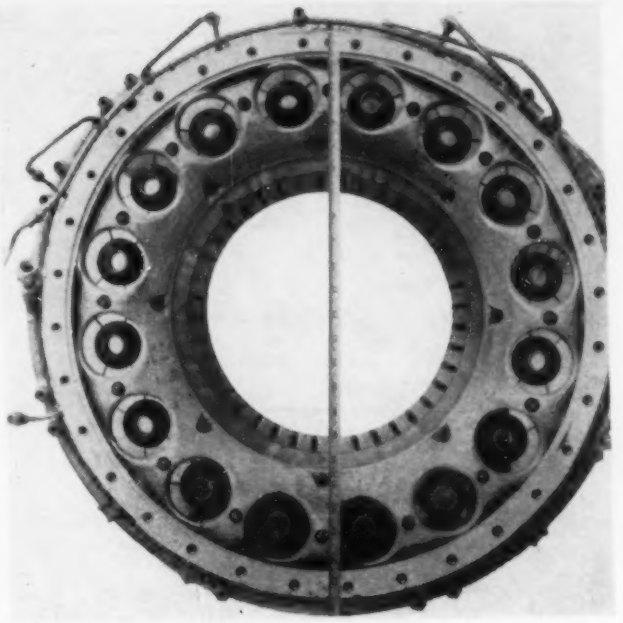
Interrogation of the BMW engineers revealed that, while they considered the combustion chamber just described as a satisfactory initial model, they were still dissatisfied with many of its characteristics – notably, the circumferential temperature distribution which apparently still could stand improvement. Another defect was the occasional complete carbonization of the space between the burner cone and the little internal cone surrounding the fuel spray. (See Fig. 9.) This was apparently an erratic occurrence for



■ Fig. 10 – Annular combustion chamber for BMW-003A



■ Fig. 11 – Temperature spread after nozzle



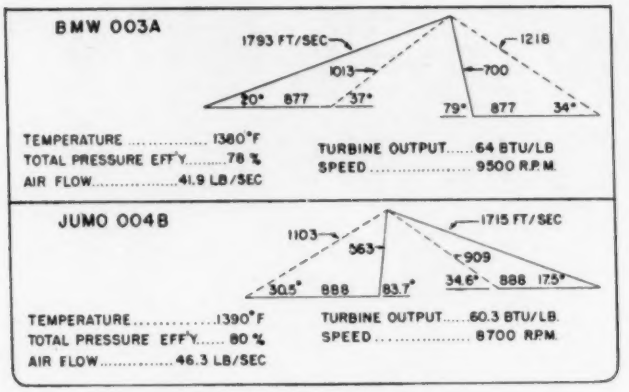
■ Fig. 12 – Inlet end of combustion chamber

■ Fig. 13 – Upstream view into combustion chamber

which no explanation was available except that it may have been the result of incorrect fuel spray cone angle. The blowout characteristics at altitude were unsatisfactory and it was not possible to restart the engine above 12,000 ft.

■ Turbine Performance

In order to minimize the risk of failure with the turbine design, both the BMW and the Junkers engineers selected single-stage turbines for their early models. The low compressor pressure ratio of these machines was partially the result of this decision. The first experimental designs used almost pure impulse blading, but the production models of



■ Fig. 14 – Turbine vector diagrams
—— Velocity relative to stator, fps
- - - - Velocity relative to rotor, fps

both the 003A and the 004B were of the partial reaction type, with about 20% reaction at the design point. Approximate vector diagrams for both units are shown in Fig. 14.

The turbine efficiency of the 003 turbine is shown in Fig. 15. The two lower curves represent the design curve and the actual test results. Evidently, the degree of reaction present in the final machine was greater than the design had anticipated, as indicated by the fact that peak efficiency occurred at a higher velocity ratio. This was undoubtedly disappointing to the Germans as it dropped the efficiency in the operating range about 1.5%. The value of u/c used here again reflects German presentation methods not universally employed in this country. The quantity u is the familiar pitch line velocity of the buckets, but c_o is the theoretical velocity equivalent to the total isentropic energy drop of the stage, rather than the nozzle spouting velocity.

The two curves just discussed were based on the static discharge pressure, while the high dashed curve is an estimation of the efficiency values obtained on the basis of total pressure drop, that is, assuming that the leaving velocity is completely useful in the jet nozzle.

■ Miscellaneous Operating Characteristics

In addition to the preceding principal performance characteristics, the 003 engine had the following miscellaneous characteristics which are of interest:

a. Combustion starts between 800 and 1200 rpm but starter engine remains engaged to 2000 rpm. Idling speed is 3000 to 3500 rpm.

b. Acceleration – 3500 rpm to 9500 rpm in 11 sec.

c. Fuel nozzle flow tolerances were ±2%, 40 atmospheres pressure.

d. Fuel pressure – gasoline: 25 atmospheres; diesel oil: 40-50 atmospheres; J-2 fuel: 50-56 atmospheres.

e. Cooling air to turbine blades and stator blades was approximately 2% of total airflow.

f. Vibration of the hollow turbine blades was an early development trouble but was alleviated by the addition of the internal air deflector and damper.

g. Turbine wheel life was 60-80 hr, although some wheels lasted 120 hr before blade failures. Design life based on blade growth was 300 hr.

h. Fundamental shaft vibration frequency was above operating speed.

i. Man-hours to manufacture – approximately 500.

j. Application:

Production airplanes – He-162 fighter.

Experimental airplanes –

Arado 234C light bomber.

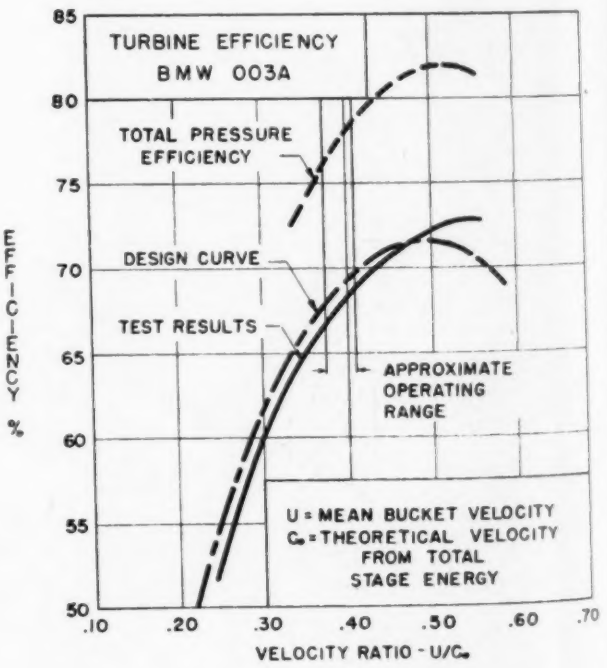
Ju-287 medium bomber.

Me-262 fighter and light bomber.

Me-264 heavy bomber.

■ Conclusion

As mentioned before the two models discussed herein were not the last word in German design but were actual production machines, and as such were somewhat crude in the opinion of the research minded engineer. However, the wealth of test data obtained during their development was of inestimable value for the practical design and performance information provided. Naturally this experience was incorporated in the later designs and the many detailed reports on the BMW-018 and 028, the Jumo 012 and 022, and other purely design models reveal how the German engineers planned to overcome the defects of the earlier machines.[1] Since the Germans were considerably advanced in the gas turbine field, it is beneficial to any engineer working on similar projects to peruse the available information in these captured documents as the vicarious experience so obtained may help him to avoid some of the less obvious pitfalls surrounding these so-called simple machines.



■ Fig. 15 – Turbine efficiency

[1] BMW-018 (large turbo-jet) – 7600 lb of static thrust; BMW-028 (propeller-turbine) – 10,000 hp at 500 mph; Jumo 012 (large turbo-jet) – 6100 lb. of static thrust; and Jumo 022 (propeller-turbine) –8000 hp at 500 mph.

FUEL SENSITIVITY and ENGINE SEVERITY in Aircraft Engines

by S. D. HERON

Ethyl Corp.

THE relation of engine severity and fuel sensitivity is discussed by Mr. Heron, who indicates that in severe engines having high cylinder and high mixture temperatures there are no superfuels and that the ceiling of lean performance number in such engines is about 100 without lead and about 150 with 4 cc of lead, whereas in mild engines performance numbers of 200 to 300 with 4 cc of lead are possible.

Virtually all of the less than 300 possible hydrocarbons within the aviation gasoline boiling range (olefins excluded), the author reports, have been tested for performance number and evaluated for temperature sensitivity. This program, he says, has been sufficient to show that there are no sound grounds for the hopes of engine builders and users for superfuels that are greatly superior to present types under still more severe engine conditions than those now in use. A decision should be made soon in choosing between the severe or the mild engine, says Mr. Heron, to guide fuel producers in the manufacture of appropriate gasoline.

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UP to about the time of Pearl Harbor aircraft-engine constructors and users in this country preferred fuels which were relatively insensitive to engine temperature. This preference was sound to a large degree, and was based upon experience going back to 1928, which showed that while benzol was an effective knock-reducing fuel component in water-cooled engines it was relatively ineffective in aircooled engines. The aircooled engines of 1928 were very severe on fuel in comparison with the aircooled types of 1942; this was largely due to cylinder cooling. Furthermore, the test methods used in full-scale engines up to about 1935 were such as to penalize sensitive fuels by rating almost entirely in terms of lean mixture performance. These fuel test methods almost completely ignored rich mixture performance, although the importance of rich performance was recognized abroad.

At about the time of Pearl Harbor it became evident in this country as a result of service trouble, that both lean and rich mixture properties of fuels must be controlled and that fuels having the same rich and lean ratings (grade 100/100) or approximating thereto were not satisfactory for the then current service engines. The result of this finding was the almost immediate specification of a rich rating for the fuel then known as "100 octane." As a result of the specification of grade 100/125 a great deal of emphasis was placed on rich ratings and this led to overemphasis. As a result of this over-emphasis the engine constructors and many users have become decidedly skeptical of fuels with a high ratio of rich to lean performance. Almost invariably, as the ratio of rich to lean performance of fuels increases, so does their sensitivity to temperature. Fuels with increased temperature sensitivity also are more prone to preignition, as might be expected.

The temperature sensitivity of fuels is now controlled within fairly close limits by the requirements of a lean (F-3) rating and a rich (F-4) rating in current specifications. It would appear, however, that control of the temperature sensitivity of fuels is only half the problem and that the temperature severity of engines should likewise be given serious consideration. The performance number of a temperature sensitive fuel may be doubled as engine conditions are varied from very severe to very mild conditions. If mixture temperatures in particular and also cylinder temperatures are to increase above present practice in some aircooled engines it would appear that fuels less temperature sensitive than grade 100/130 will be desirable. On the other hand, if engine severity is to be reduced fuels more sensitive than grade 100/130 may be desirable.

In any case, from the standpoint of national security, it would seem that the joint problems of engine severity and fuel sensitivity should be coordinated as regards military aircraft. In regard to commercial aviation the problem is important but production of suitable fuel grades to fit the engines in use at any particular time can readily be accomplished in view of the limited peacetime demand in relation to the present total fuel production capacity.

[This paper was presented at the SAE National Aeronautic Meeting, New York City, April 5, 1946.]

In regard to the air arms of the military services, coordination of fuel sensitivity and engine severity is important since engine design and fuel manufacturing are inevitably tied together. It is obviously unsound to design and develop engines which require fuels which in time of emergency will necessitate gigantic manufacturing resources and crude oil supplies which will not be available. Likewise, it is unsound to develop fuel manufacturing methods giving a product which will so handicap the performance of even specially developed engines that military equality cannot be obtained.

In the past neither aircraft fuels nor aircraft engines were well enough understood so that engines could be built for particular types of fuel. As a consequence (and quite soundly so) fuels were made to fit the current engines and this resulted in the adoption of 100-octane fuel based on iso-octane as the component which made it possible. It is possible that the severe type of engine using very high mixture temperature is one of outstanding military advantage and if so, it is obviously desirable to develop suitable fuels and manufacturing resources for such fuels. It is likewise possible that the milder type of engine which performs most effectively with a fuel at least as sensitive as grade 100/130 has military advantages. Some authorities think the milder type of engine has proved its advantages in World War II. The milder type of engine can make valuable use of fuels such as triptane and similar materials which are not of significant advantage in the very severe engines except at very rich mixture. All in all, it would appear that engine severity and fuel sensitivity should be coordinated in the next year or two. The question of significant routine measurement of temperature sensitivity of fuel is not completely understood as yet and measurement of engine severity is as yet almost unknown let alone expressing it on a numerical scale. In regard to aircraft-engine severity the definite knowledge does not extend much beyond the fact that engines with low cylinder and mixture temperatures give best performance on fuels such as grade 100/150, whereas engines with high cylinder and mixture temperatures do best with less sensitive fuel types.

The problem of fuel sensitivity and engine severity appears to be ripe for discussion by the military services, the engine constructors and the fuel producers. The engine constructors are not generally aware that there are less than 300 possible hydrocarbons in the aviation gasoline boiling range if olefins are excluded. Olefins are very temperature sensitive and otherwise do not appear attractive for present high performance engines and, therefore, need not be considered at this time. Engine constructors and users are likewise not aware that nearly all the high performance number hydrocarbons in the aviation fuel boiling range have been evaluated as to performance number and approximately evaluated as to temperature sensitivity. One group of hydrocarbons within the aviation fuel boiling range which has not yet been completely investigated has been sufficiently explored to determine that the group as a whole is decidedly temperature sensitive. The engine constructors and users are likewise not aware that the evaluation has shown that there are no superfuels at very severe engine conditions. Some engine constructors, who are not aware that there are less than 300 possible combinations of hydrogen and carbon within the aviation fuel boiling range (olefins excluded) and that these have been evaluated, consider that it should be possible to invent new hydrocarbons to fit engine requirements. Some engine constructors visualize fuels of high performance number, excellent volatility, and almost no temperature sensitivity and are unable to believe that such fuels cannot be invented as a new piston is invented to suit a need.

Many engine designers and engine users are under the impression that lean and rich performance numbers of 200/250 are in sight and are not aware that while such lean and rich performance numbers can be obtained they are only possible with reasonably mild engine conditions. That the ceiling of lean performance number with severe engine conditions is about 100 with no lead and about 150 with 4 cc of lead is certainly not clear to many in the engine field.

It would seem that the achievement of synthesizing and evaluating practically all the hydrocarbons (olefins excluded) within the aviation fuel boiling range should be brought to the attention of the informed aviation fuel consumers. Synthesis and evaluation of practically all the possible desirable components of aviation fuels is an achievement of something more than major proportions. The major credit for this achievement is due to the oil industry and its foresight and breadth of vision which resulted in the formation and continued support of API Project 45 (formerly API Hydrocarbon Research Project), which has been the major contributor to the present state of knowledge. Among other outstanding contributors have been the National Bureau of Standards working on a project financed by NACA and the NACA itself at the Aircraft Engine Research Laboratory, Cleveland. It would seem to the writer that this achievement is worthy of publicizing in technical circles.

The general tendency of fuel manufacturing and fuel research in this country is in the direction of fuels more sensitive than iso-octane and this would appear to be good reason for an early program of coordination of engine severity and fuel sensitivity. If the tendency to sensitive fuels is unsound from the standpoint of national security it would appear that this should be determined as soon as possible.

This note appears to put considerable emphasis upon the hopes of the engine constructors for superfuels, which hopes are almost impossible of achievement. The petroleum industry, or at least some sections of it, has likewise failed to comprehend completely the nature of the joint fuel-engine problem. Part of the petroleum industry has publicized superfuels without realizing that such superfuels could only be superfuels in engines which might be quite undesirable for aircraft use. Some of the discussions of superfuels have ignored the fact that, while for a given power output superfuels would enable engine size to be greatly reduced, it would be at the expense of heat exchangers which might be as large as the engine.

In conclusion, it may be remarked that if all fuels had the same temperature sensitivity there would be no performance-number (or octane-number) controversy either technical or commercial. Alternatively, if all engines had equal severity there would be no octane-number controversy and an accurate rating in any one engine would be valid for all engines with, of course, the proviso that fuel distribution effects within the fuel-air induction system of multicylinder engines would modify the validity of this one rating.